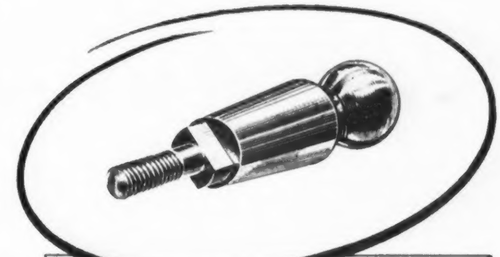
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 43 No. 567 JUNE, 1953 PRICE: 3s. 6d.

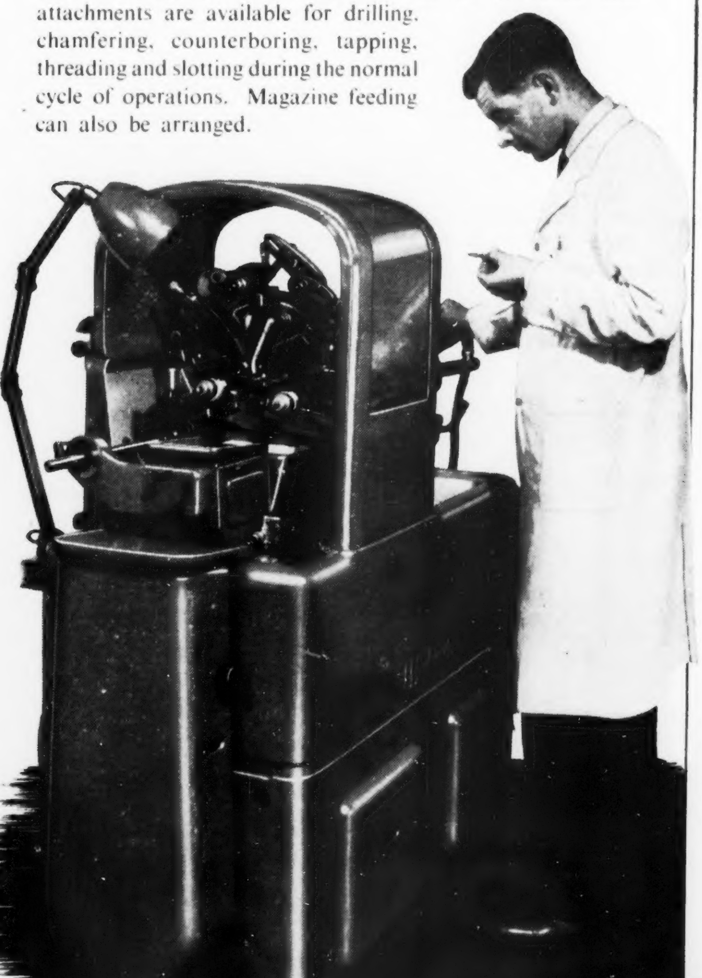
Wickman Sliding Head Precision Automatic

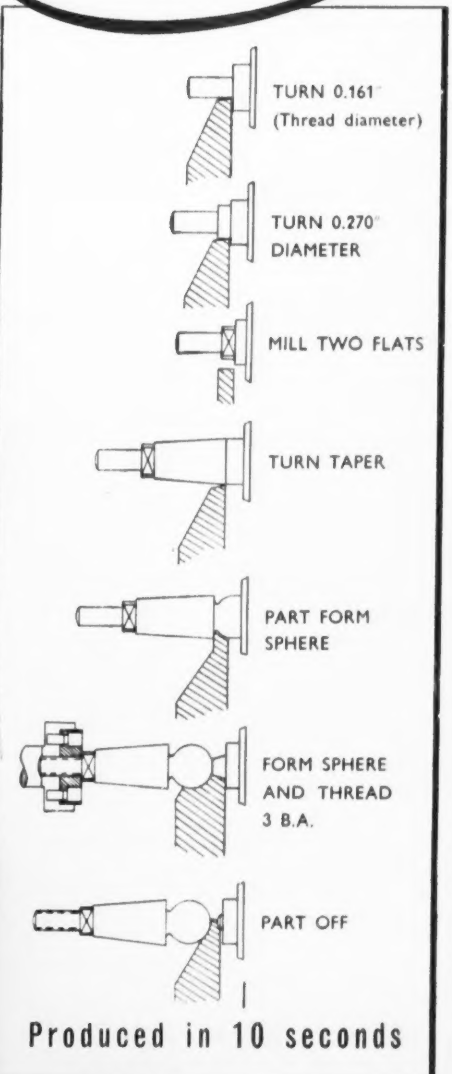
$\frac{7}{16}$" CAPACITY



Primarily designed for the production of small components for the clock and precision instrument industries, this automatic is now recognised as one of the most efficient machines for general small turning work in large quantities.

Its extreme versatility is well demonstrated in the accompanying example. Here taper turning, threading, flat generating and form turning are all accomplished in a single cycle of operations, and the part is produced complete from bar stock in 10 seconds. Multiple diameters, pivot points at one or both ends, back shoulders, tapers, bevels, etc., can all be produced and attachments are available for drilling, chamfering, counterboring, tapping, threading and slotting during the normal cycle of operations. Magazine feeding can also be arranged.





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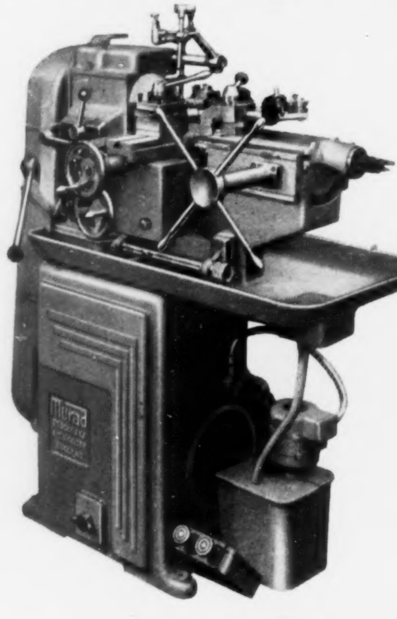

Murad

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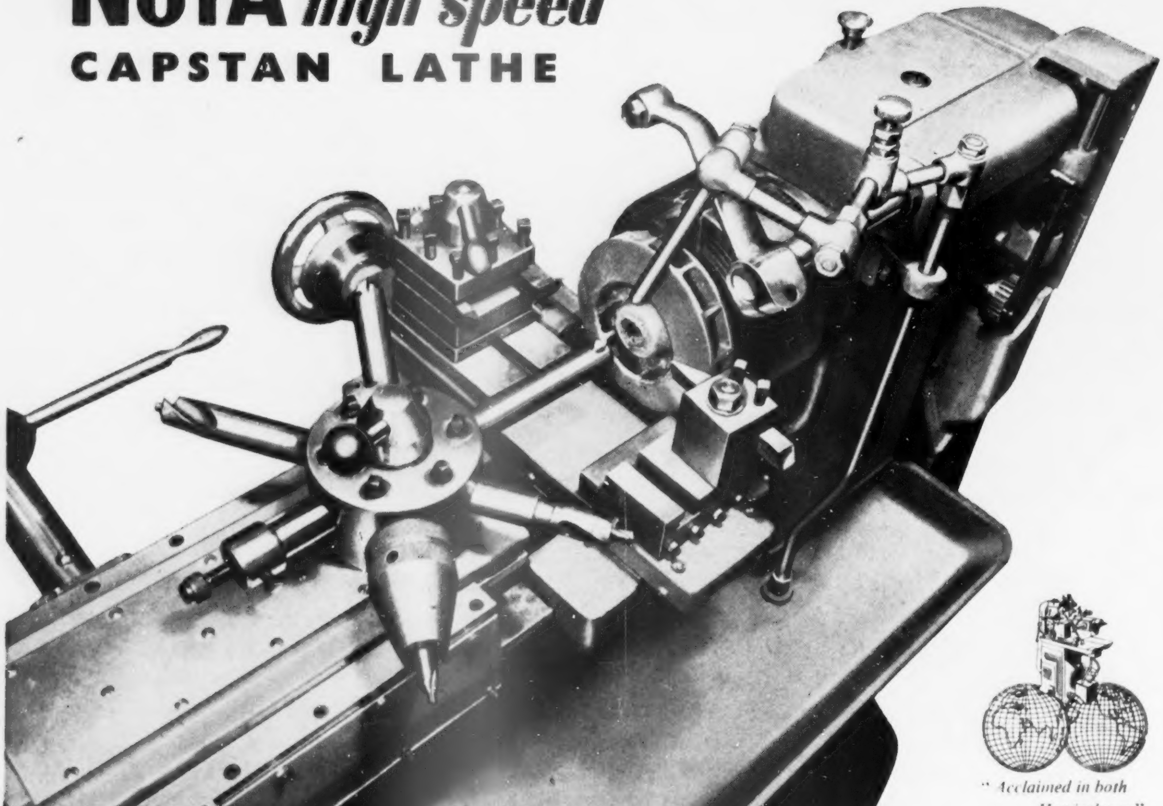

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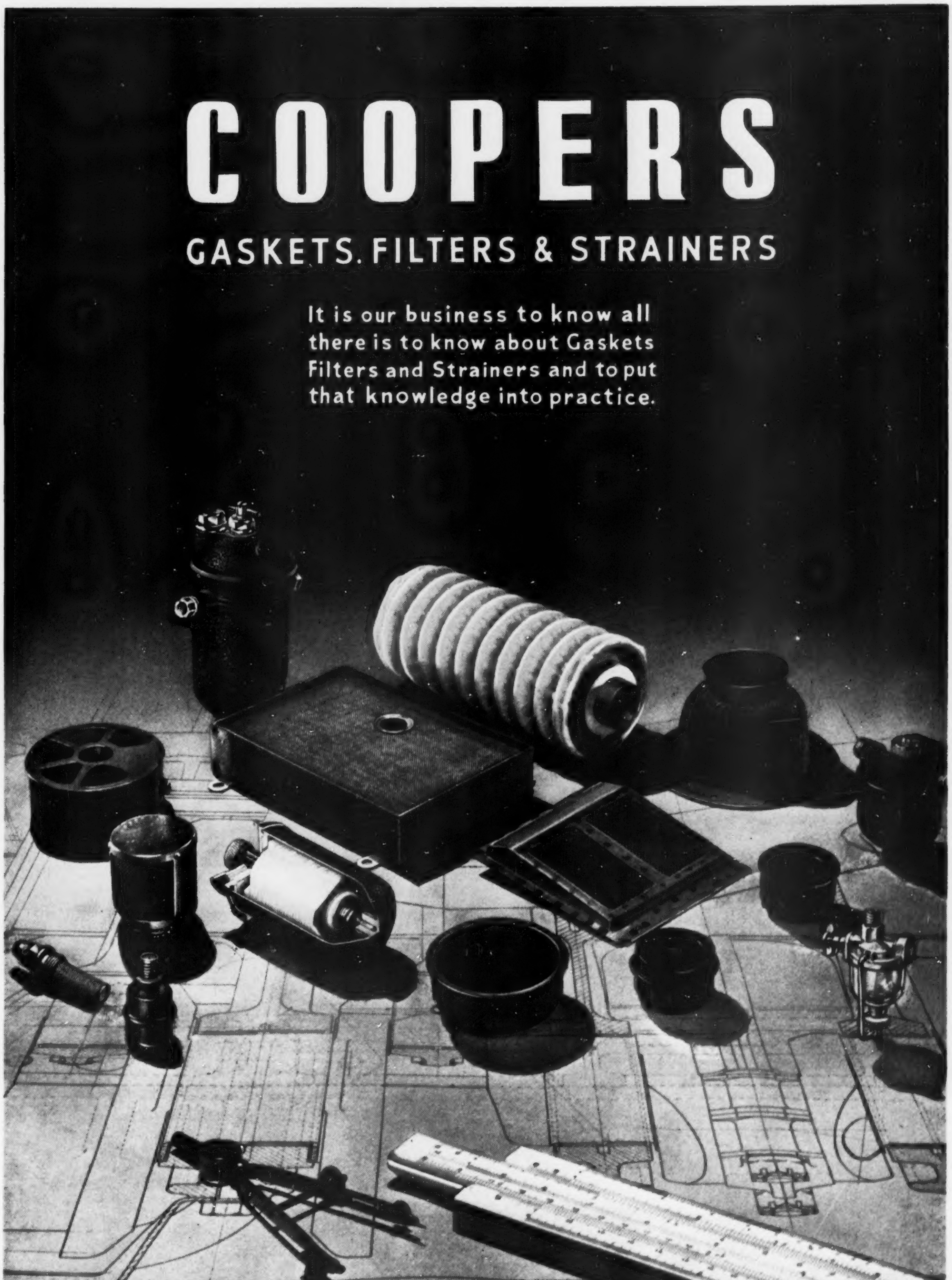
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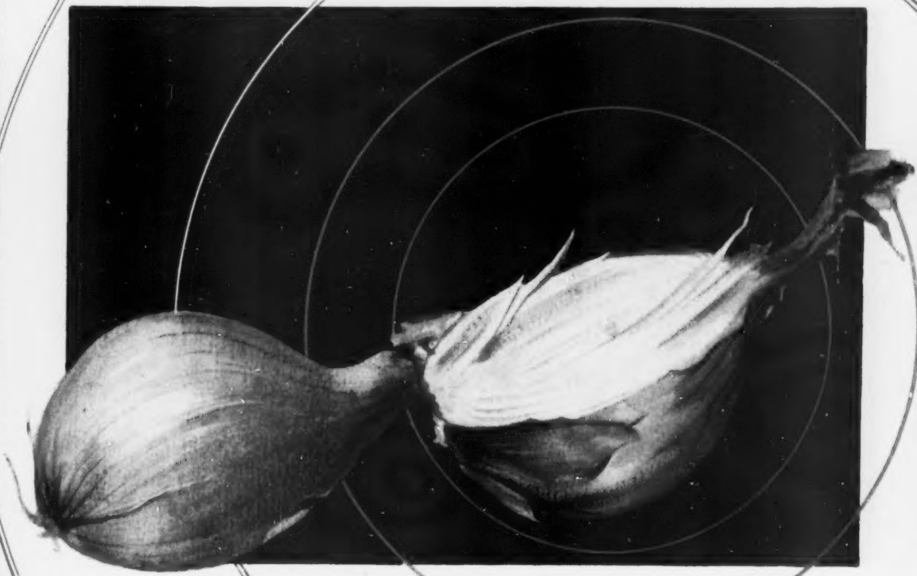

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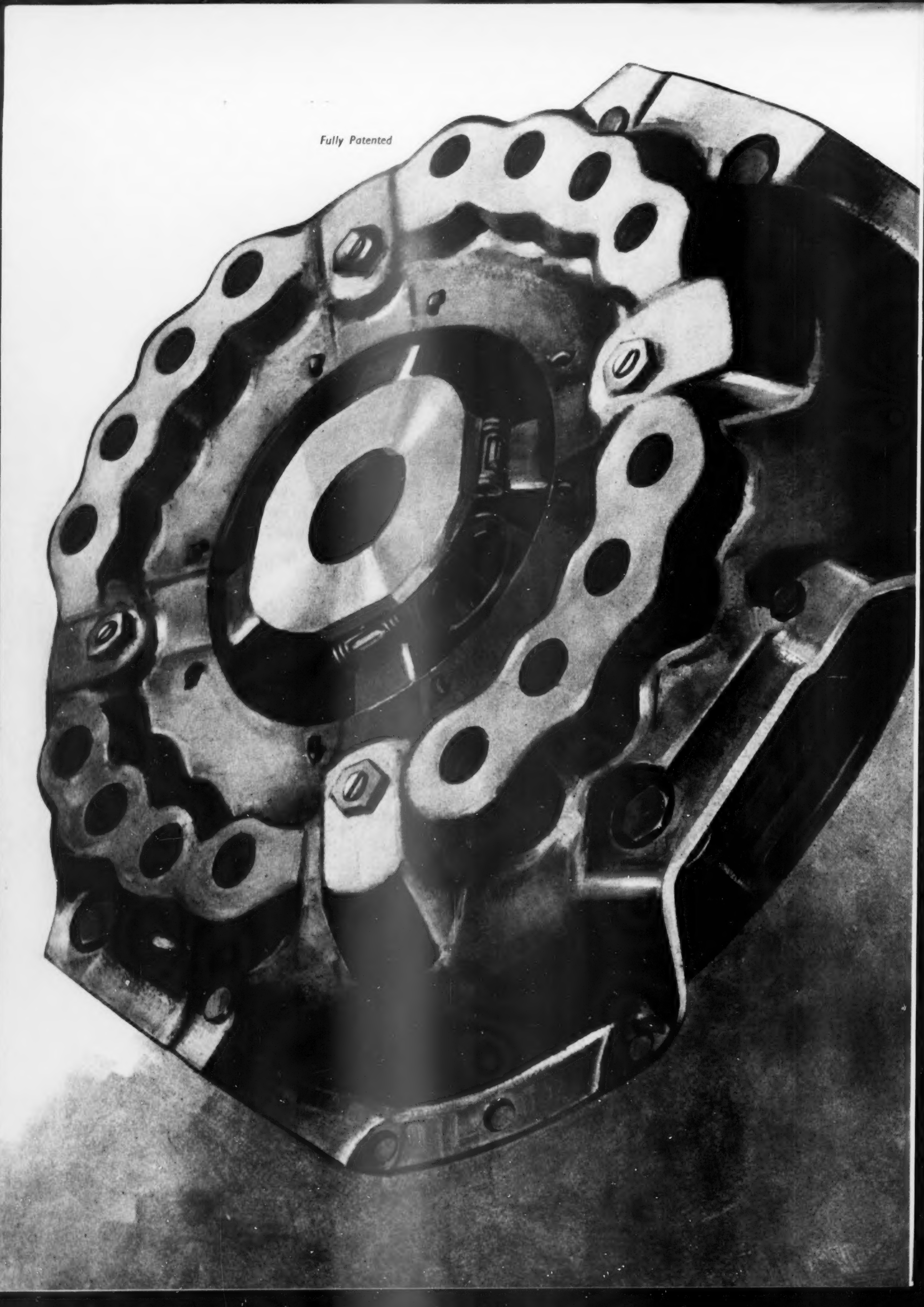
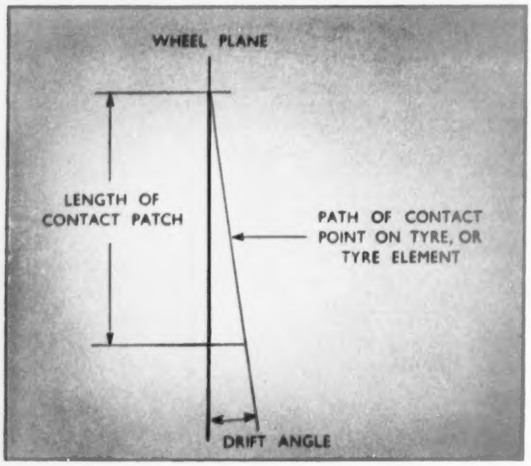


SUBTLETIES OF STEERING

Causes of over- and under-steer: 2. Tyre properties (a)

IN this survey of tyre properties we are only concerned with one side of the eternal triangle of tyre behaviour comprising the struggle between cornering power, enveloping or cushioning power, and durability; our subject is cornering power. Every tyre has some sideways flexibility; more when it has a full tread than when it has worn down, more when it is inflated to a low pressure than when it is blown up hard. These are points we can easily verify for ourselves if we doubt our own reasoning powers. This sideways



The text explains how both thrust and torque are necessary to make the 'equator' of a tyre tread take up this shape, whether the tyre is stationary or rolling.

flexibility is the reason why tyres drift sideways when they are subjected to a sideways load while rolling along.

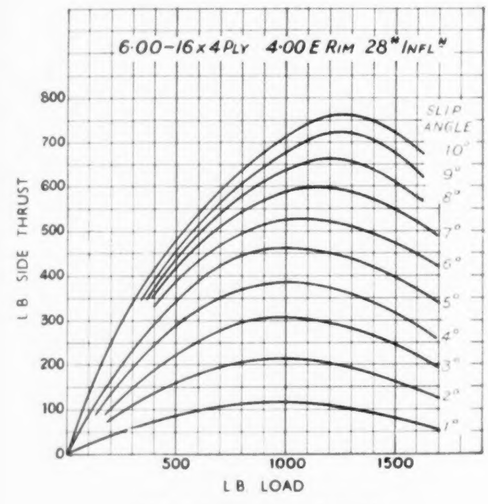
Suppose for the sake of argument that the contact length of a tyre comprises ten 'elements', and that each element has a sideways stiffness of 200 lb. per inch. A total sideways load of 500 lb. will give, when the tyre is stationary, a load of 50 lb. per element or a total sideways deflection of one quarter of an inch.

Now suppose we apply a torque about a vertical axis, through the centre of the contact patch, it is obvious that the result will be an angular deflection about this axis. Suppose the length of the contact patch is 9″ (i.e. an inter-element spacing of 1″) and that we apply a torque just sufficient to give no deflection of the end element in one direction, and therefore a deflection of $\frac{1}{4}$″ of the other end element. The angle turned through is $\frac{0{\cdot}25}{4{\cdot}5}$ radians . . . about 3·2°. The torque required can be arrived at by taking the relative deflection of each element, finding the corresponding load to produce that deflection and taking its moment about the axis. The sum of all these moments gives the torque. Without boring you with the details of the calculation, this means a torque of some 76 lb.-ft.

If now we start to roll the wheel along so that each element of tyre, as it arrives at the ground, is at the zero deflection end, it will be seen that to produce the required total sideways load, the elements will have to assume successively the positions we have just seen to be adopted by the elements of the stationary wheel subjected to the same sideways load and twisted so that one end element has zero deflection. The true direction of motion is then at the angle of 3·2° to the plane of the wheel, i.e. a drift angle of this amount has been developed, and a torque of 76 lb.-ft. must be applied to maintain the wheel in that position, i.e. there is a returning or self-centring torque of that amount. Another way of expressing the torque is to say that the sideways load of 500 lb. is being applied at a distance of 0·152 ft. behind the vertical through the wheel axis. For a rolling radius of 13″ this is the equivalent of a caster angle of 8°.

As the total sideways load is increased, there will come a time when the sideways load on the rear element is more than it can take without slipping sideways. As this occurs the drift angle will start to increase more rapidly with increasing sideways load, and a little further consideration will show that the effective moment arm of the sideways load will begin to decrease, i.e. the effective or apparent caster angle will then begin to diminish.

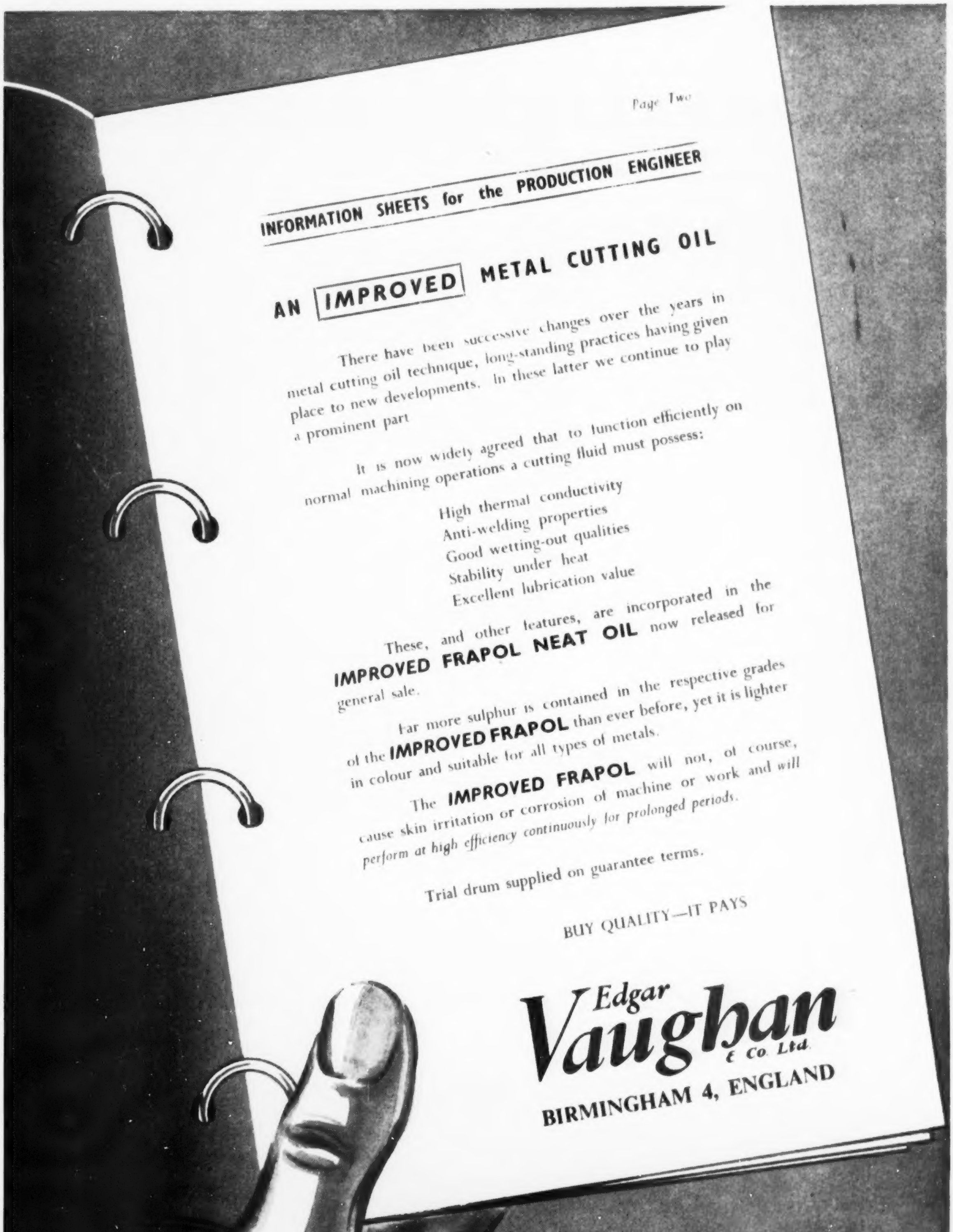
The effect of the load carried by the tyre is interesting; below the optimum, the length of the contact patch is reduced, the number of elements is reduced, the sideways rating reduced, and therefore a given sideways load will



SIDE THRUST OVER LOAD FOR VARIOUS SLIP ANGLES

The basic diagram showing how side thrust in cornering power varies with the load carried by a tyre for different drift angles.

produce a greater mean deflection; this on a reduced contact patch length will imply a greater drift angle. Above the optimum, the increased vertical deflection results in a greater sideways deflection, and although the length of the contact patch is increased, the increase in sideways flexibility is eventually greater, and again a greater drift angle is developed.


Page Two
INFORMATION SHEETS for the PRODUCTION ENGINEER
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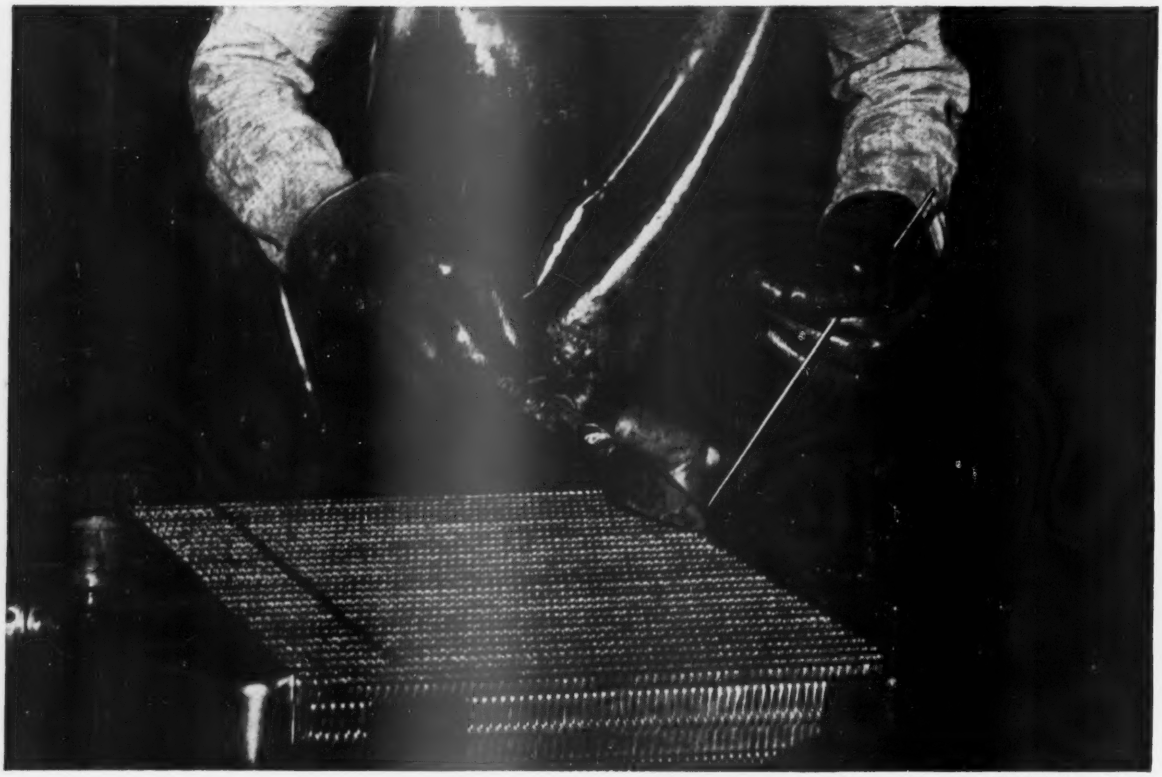

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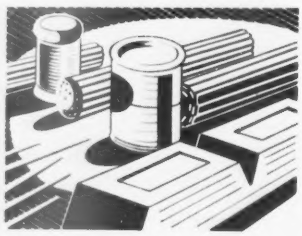

Enthoven Body Solder is supplied in sticks to standard specifications or special alloys.

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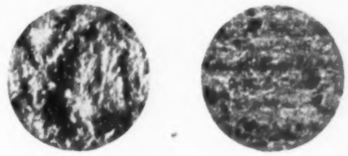

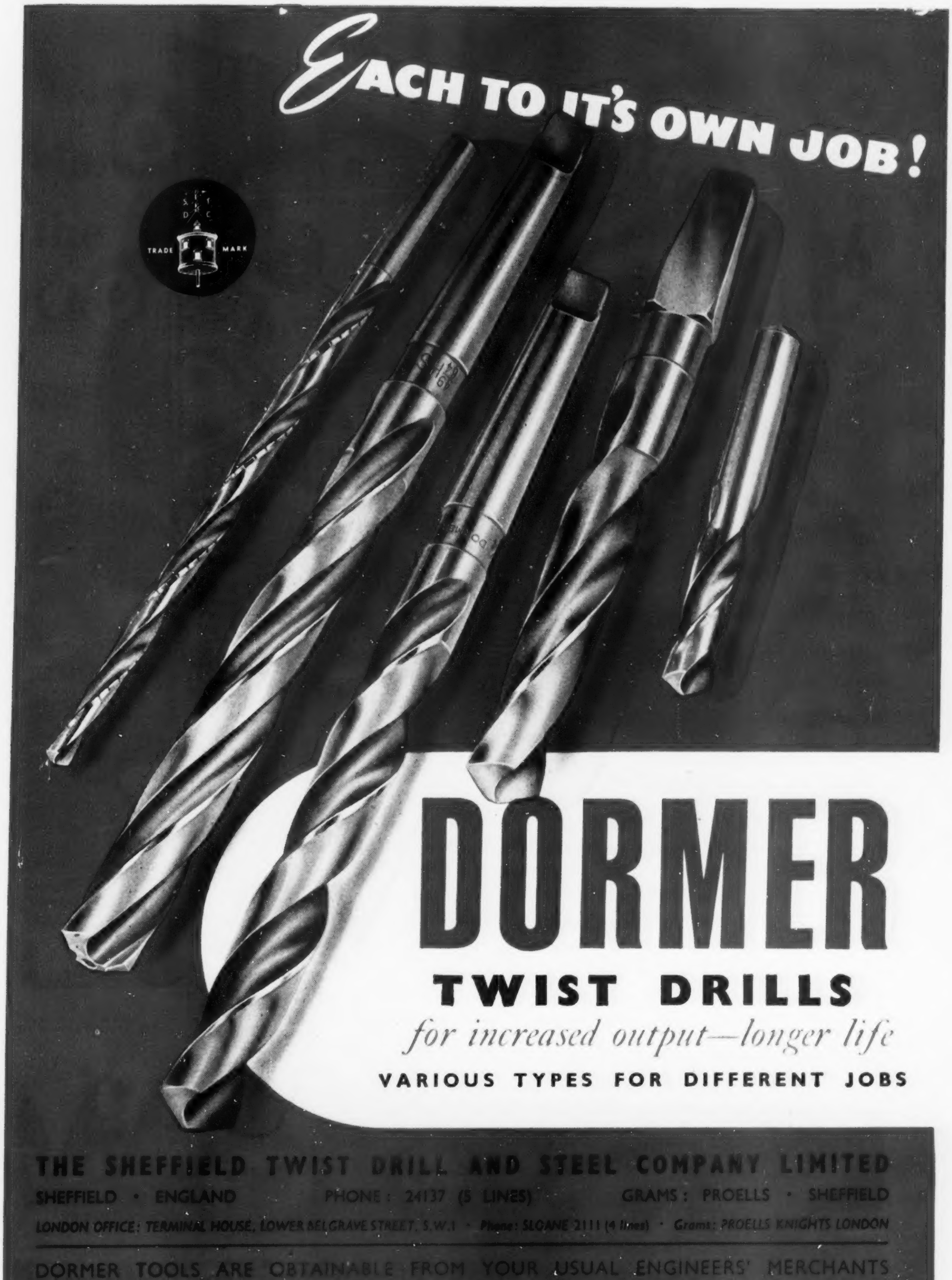

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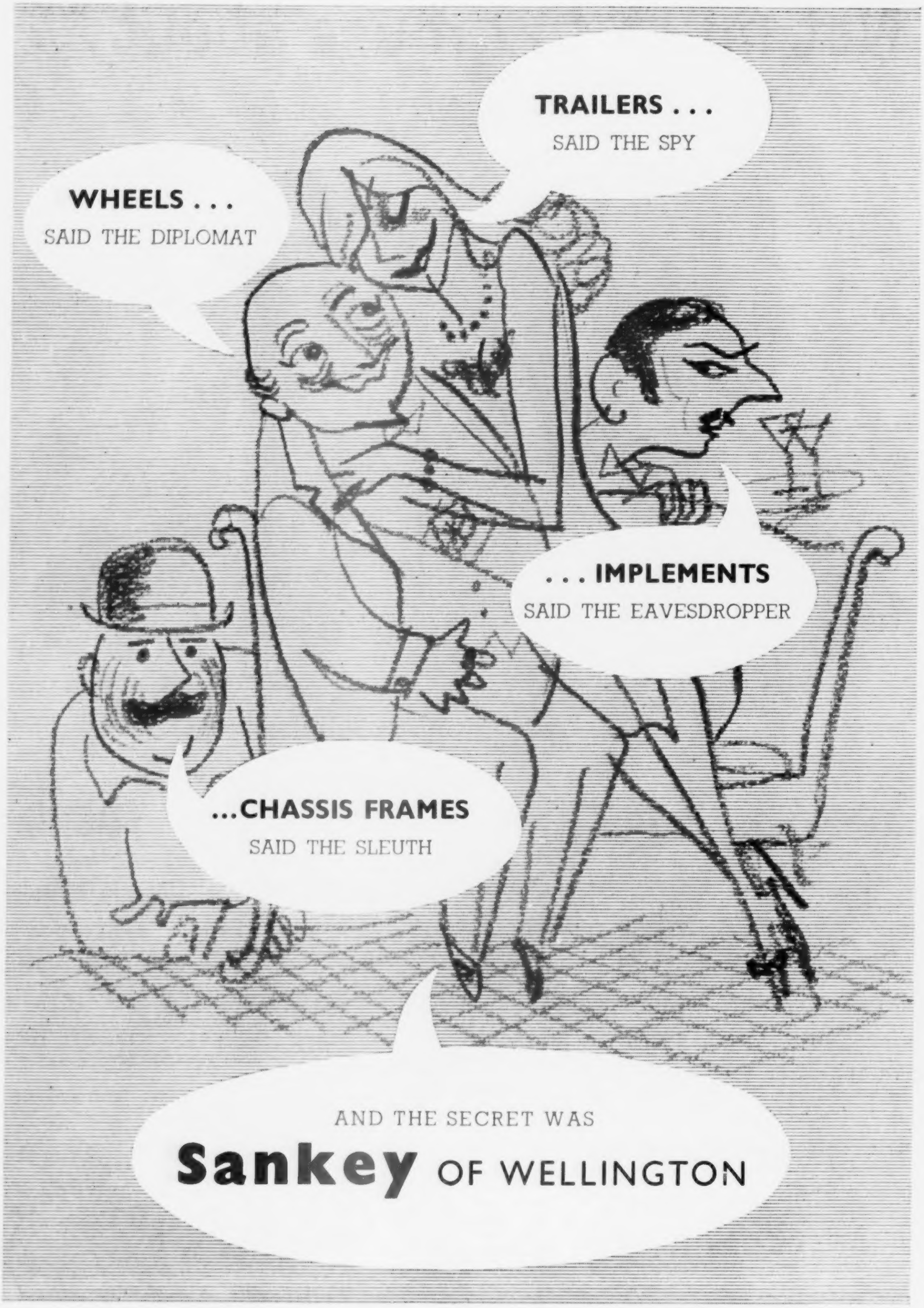

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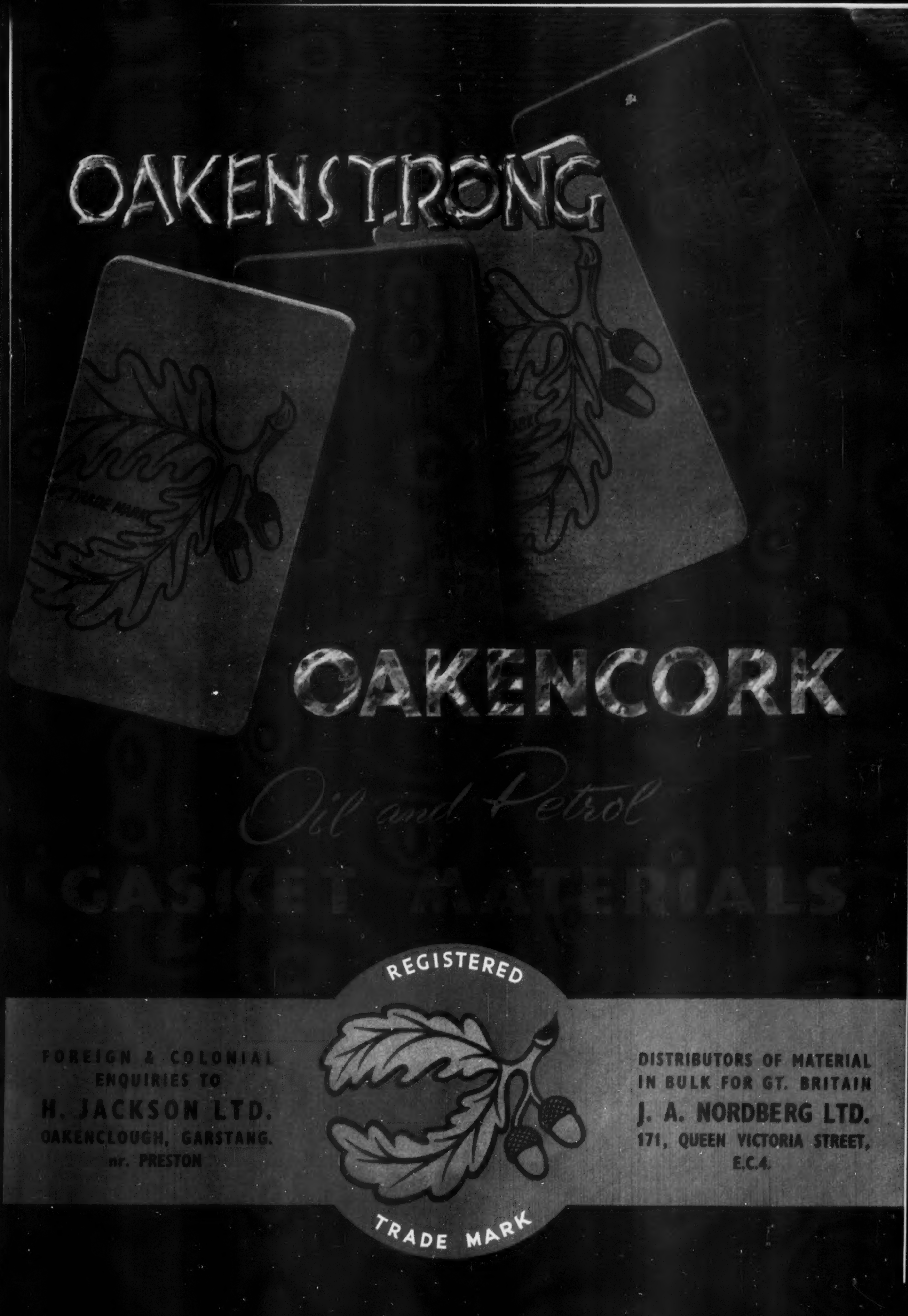
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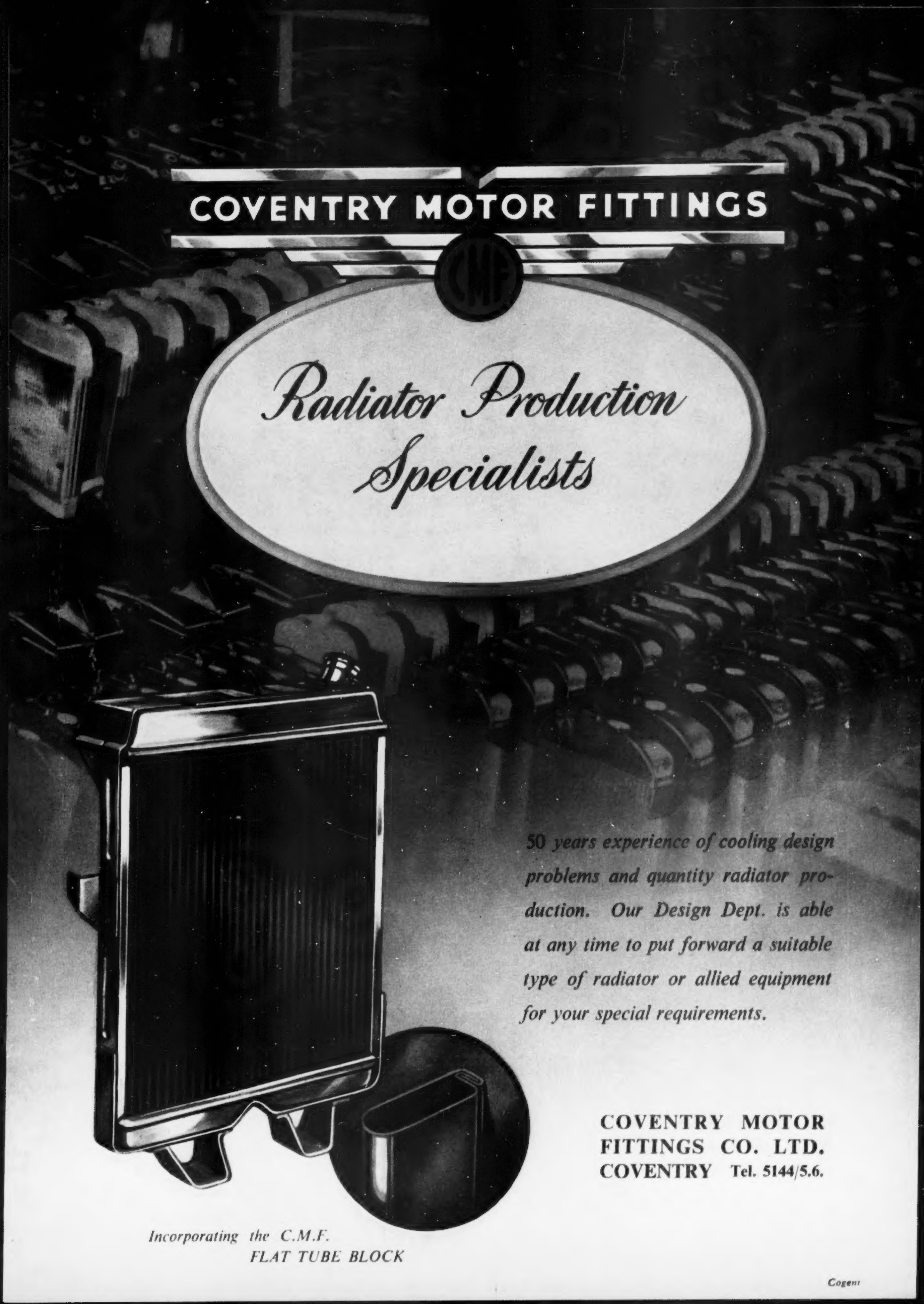
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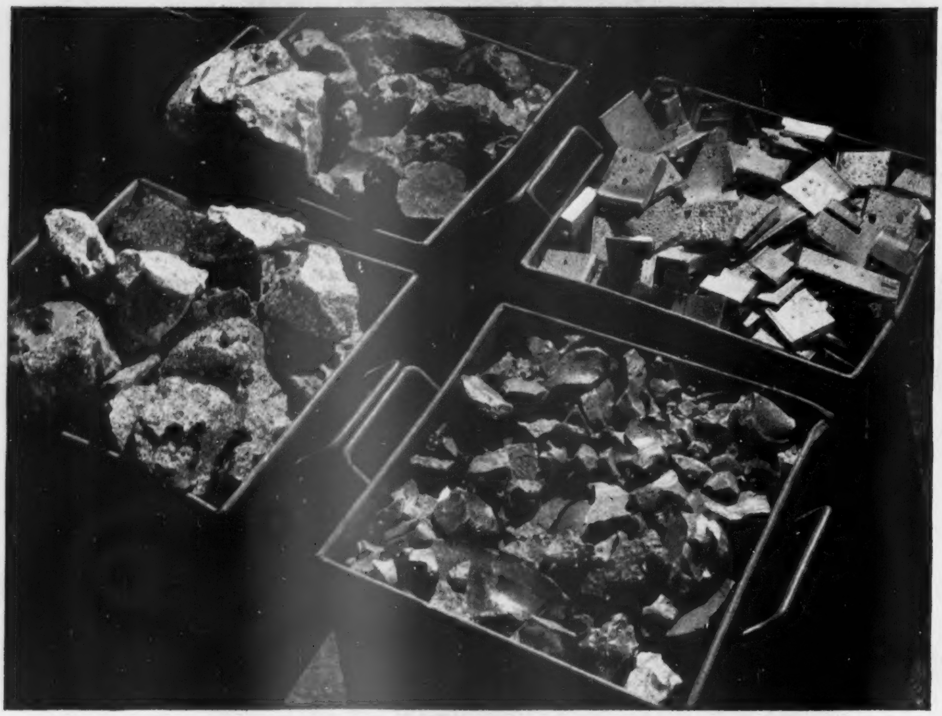

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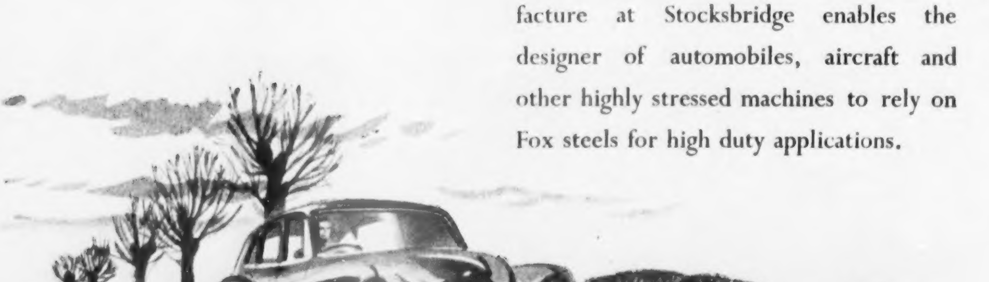

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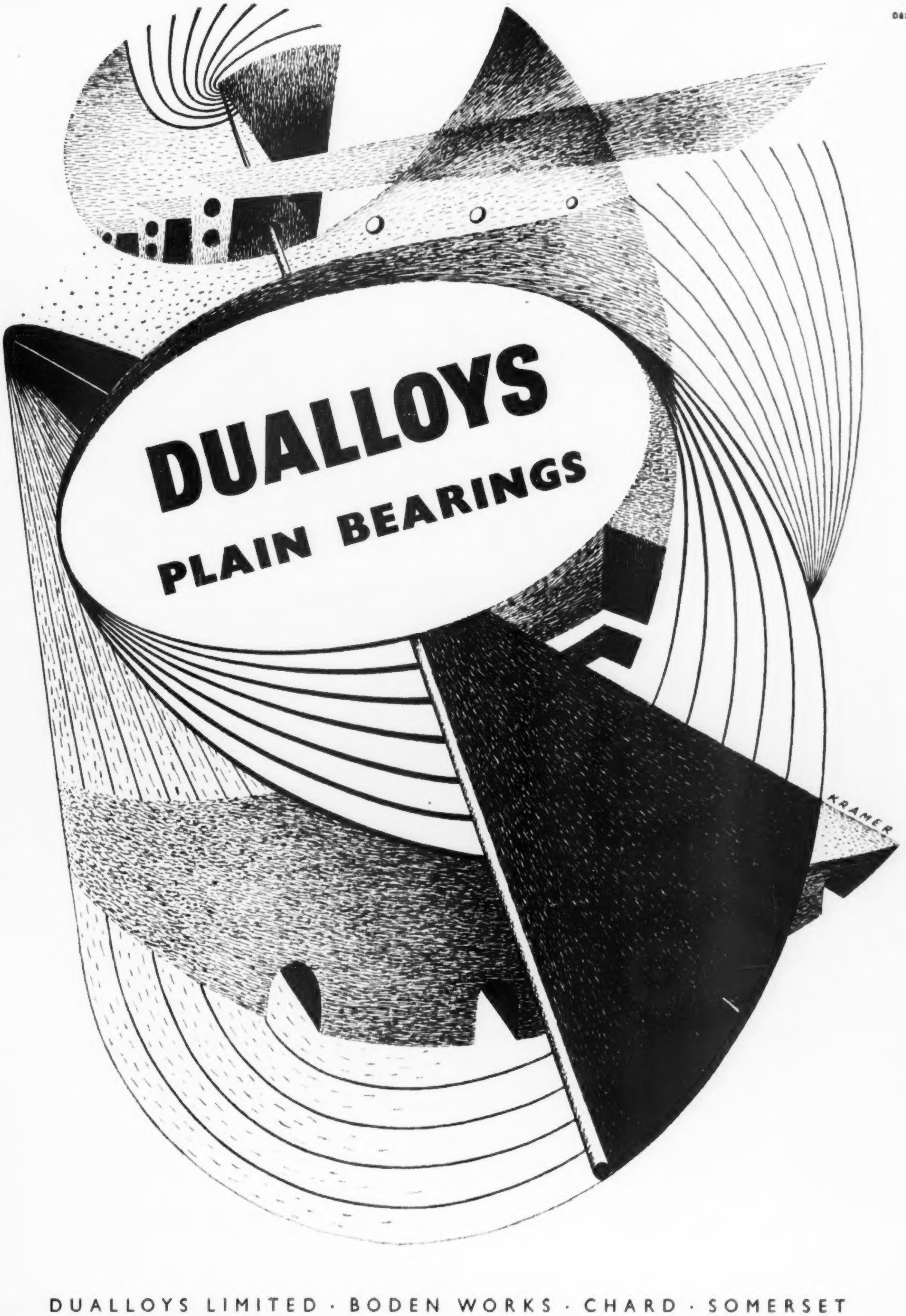
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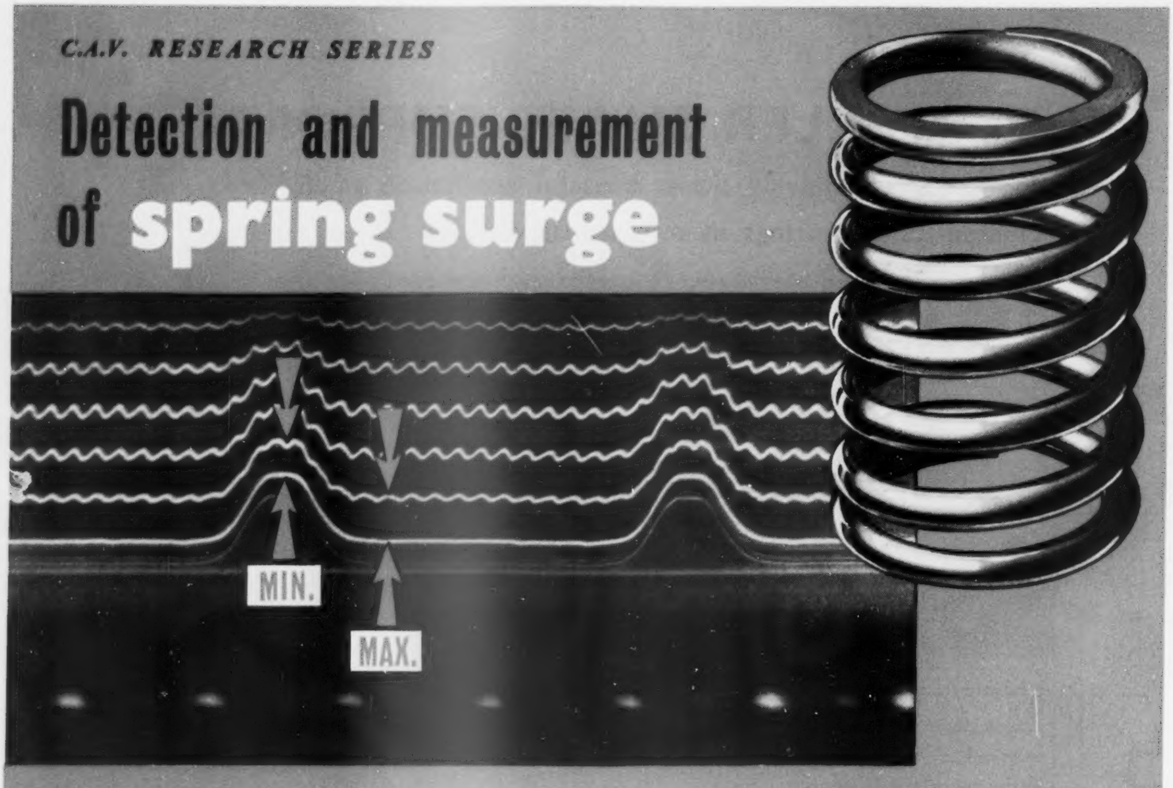
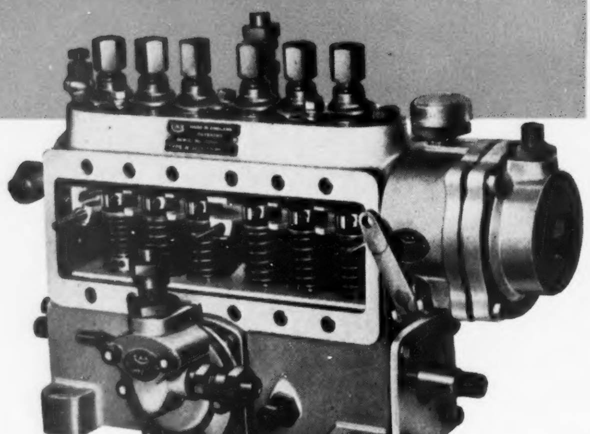


The firm line shows the cam rise over two cycles. The lower dotted trace shows time intervals of 10 milliseconds.



When a spring is abruptly compressed, its coils may be caused to "surge", that is, the centre coils are displaced relative to the ends to a greater extent than during the slow compression of the spring, and then vibrate.

Normally, the compression of the spring subjects the material to stresses which are well within its fatigue limit. When the spring surges, however, the oscillation of the coils is added to the normal movement. The stress-range to which the material is subjected is enhanced, and may become excessive.

The illustration above is from a research laboratory record of surge in a spring, the wavy lines showing the displacements of individual coils. The stress range in any turn can be deduced by observing the minimum and maximum turn-to-turn distance.

Surge is an ever-present hazard in coil springs which are compressed abruptly, such as tappet or injection pump plunger springs, and research of this type enables it to be guarded against in the design of equipment.

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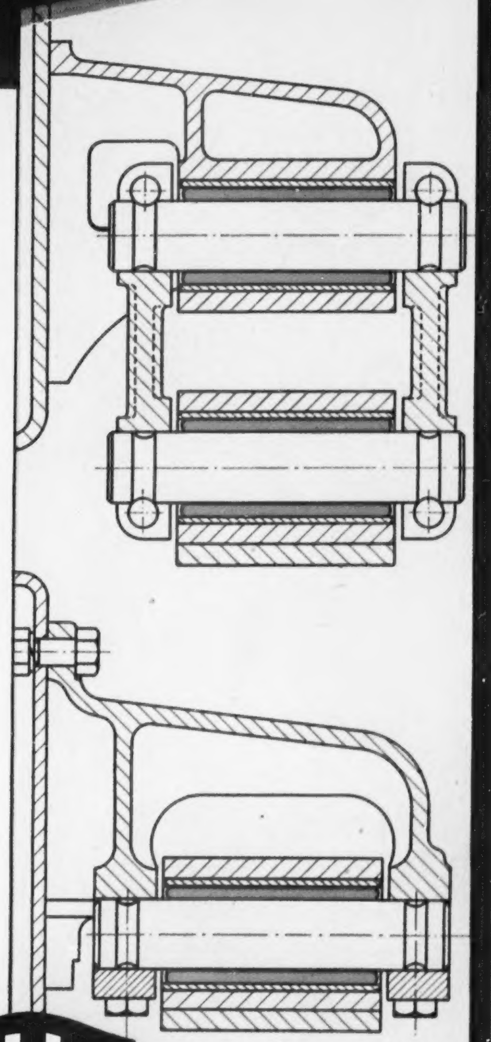
Ultra Duty shackle-pins



Metalastik solutions to the shackle-bearing problem have been widespread.

For the private car and light van, the Metalastik Ultra-Duty bush has become the nation's standard: for the heavy public service and commercial vehicle the Metalastik High-Duty shackle-pin (first evolved at the request of Messrs. Leyland), has proved to be a spectacular success, and still good for continued service after 400,000 miles. Now Metalastik extend the range and provide, for the lighter buses and goods vehicle, the **Ultra-Duty shackle-pin.** This is an entirely new development combining the outstanding features of the Metalastik High-Duty shackle-pin, the preloading technique of the Ultra-Duty bush and the famous Metalastik rubber-to-metal weld, bonding the rubber directly on to the pin.

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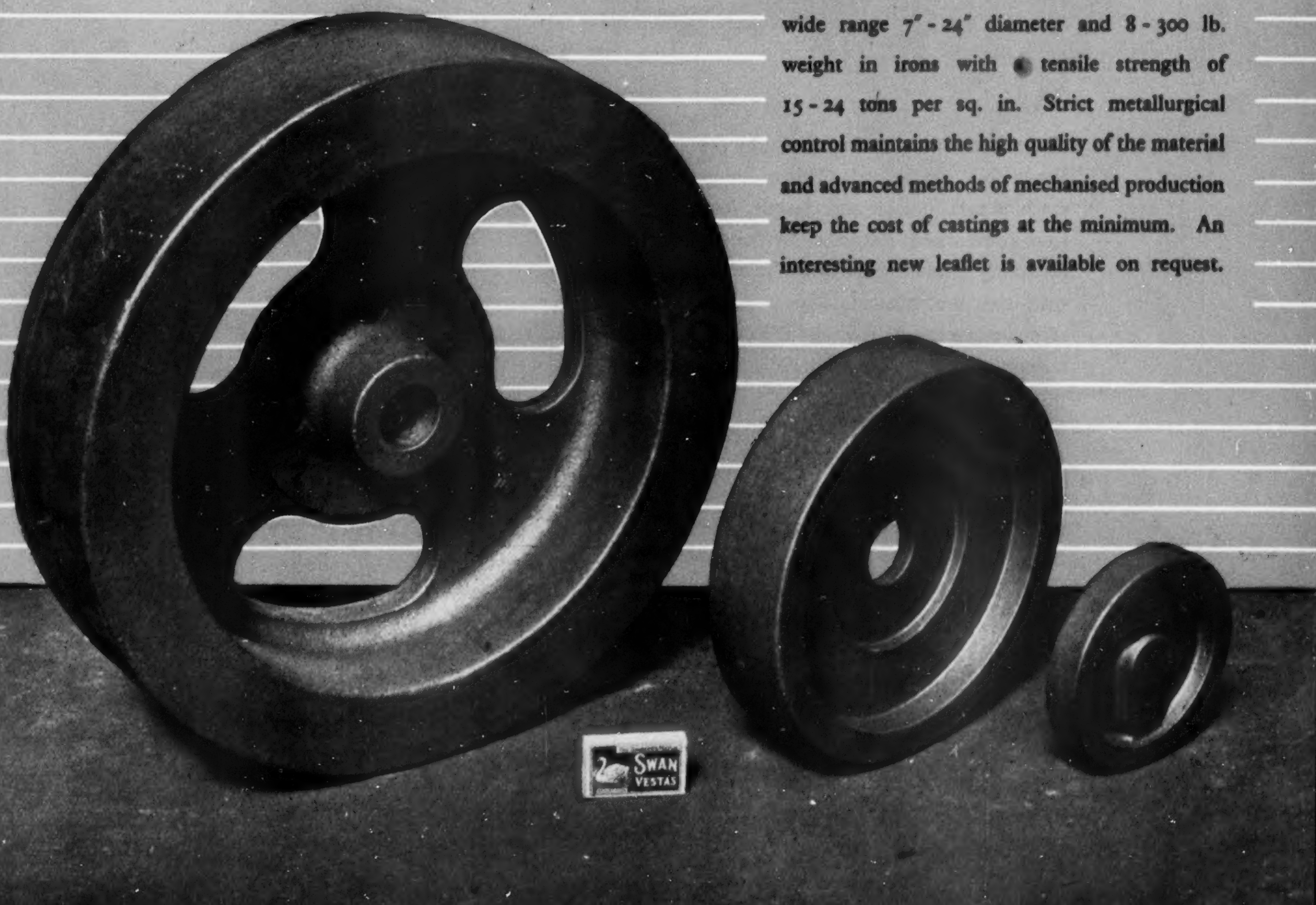
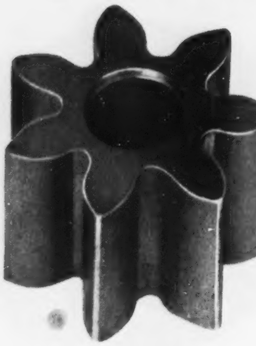
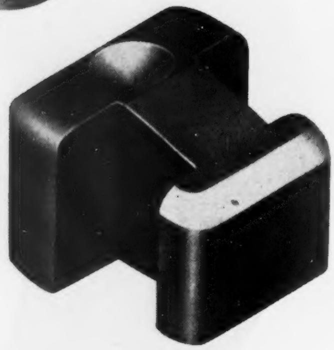
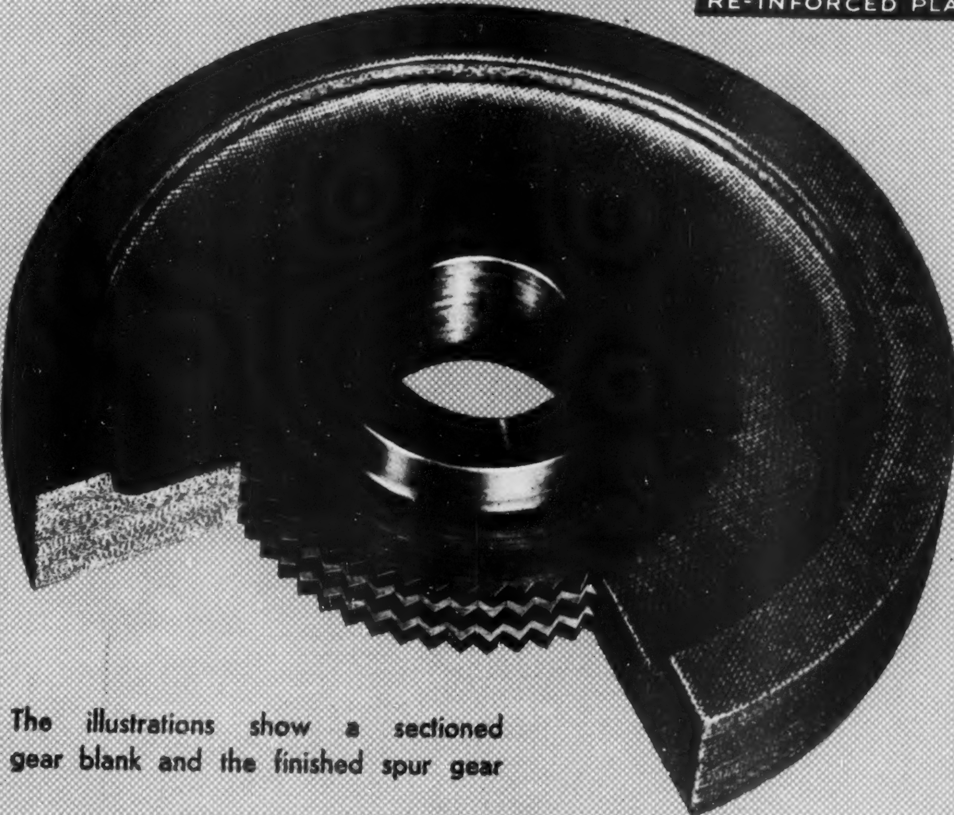


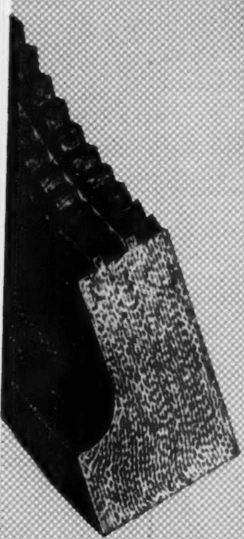
DARTMOUTH AUTO CASTINGS LIMITED

SMETHWICK 40 BIRMINGHAM

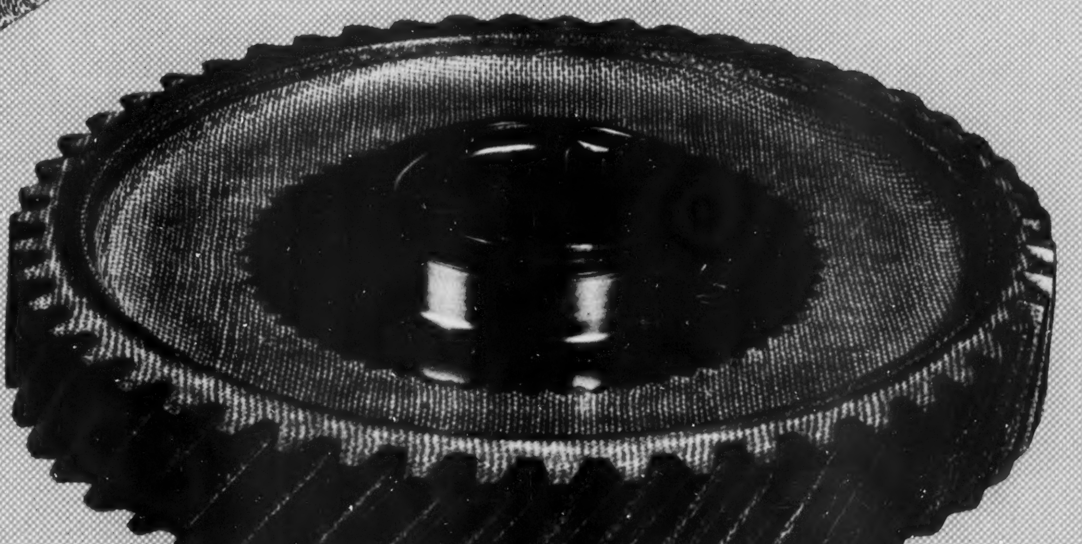






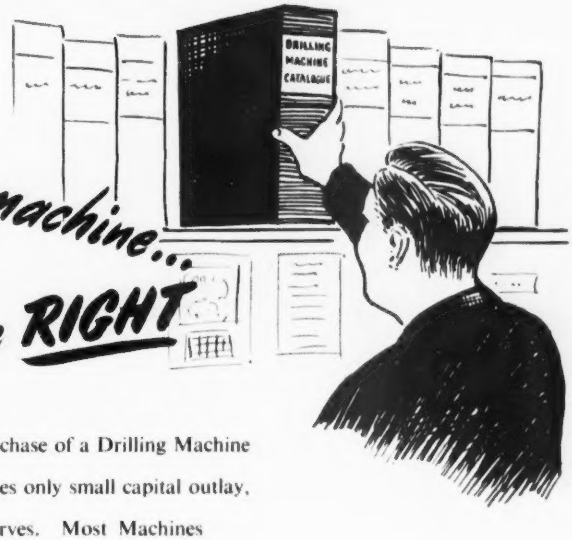


The illustrations show a sectioned gear blank and the finished spur gear



Illustrations by courtesy of Morris Motors Limited

It's easy to buy a drilling machine...



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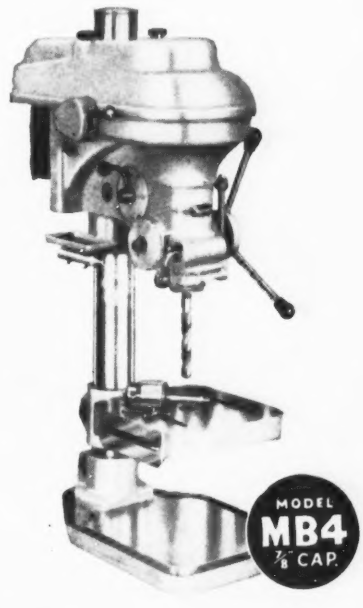

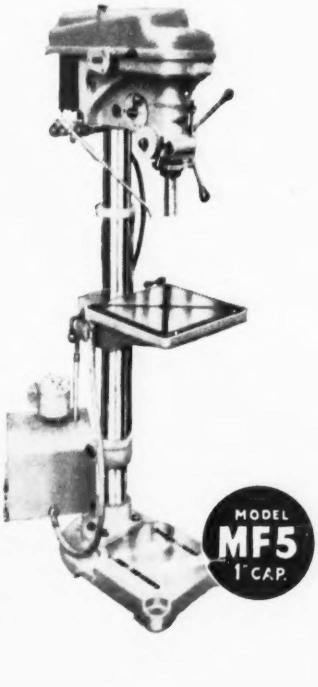

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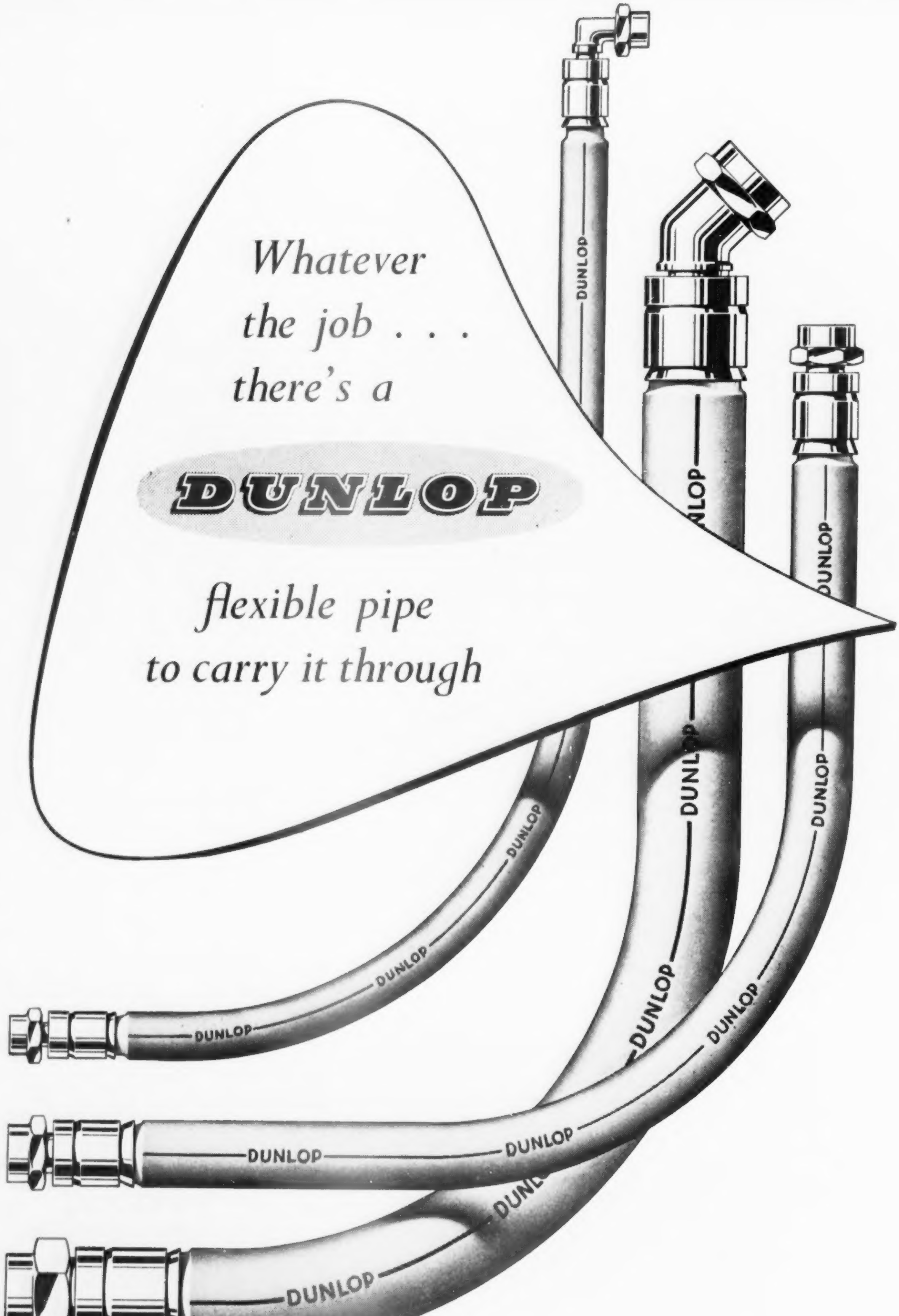
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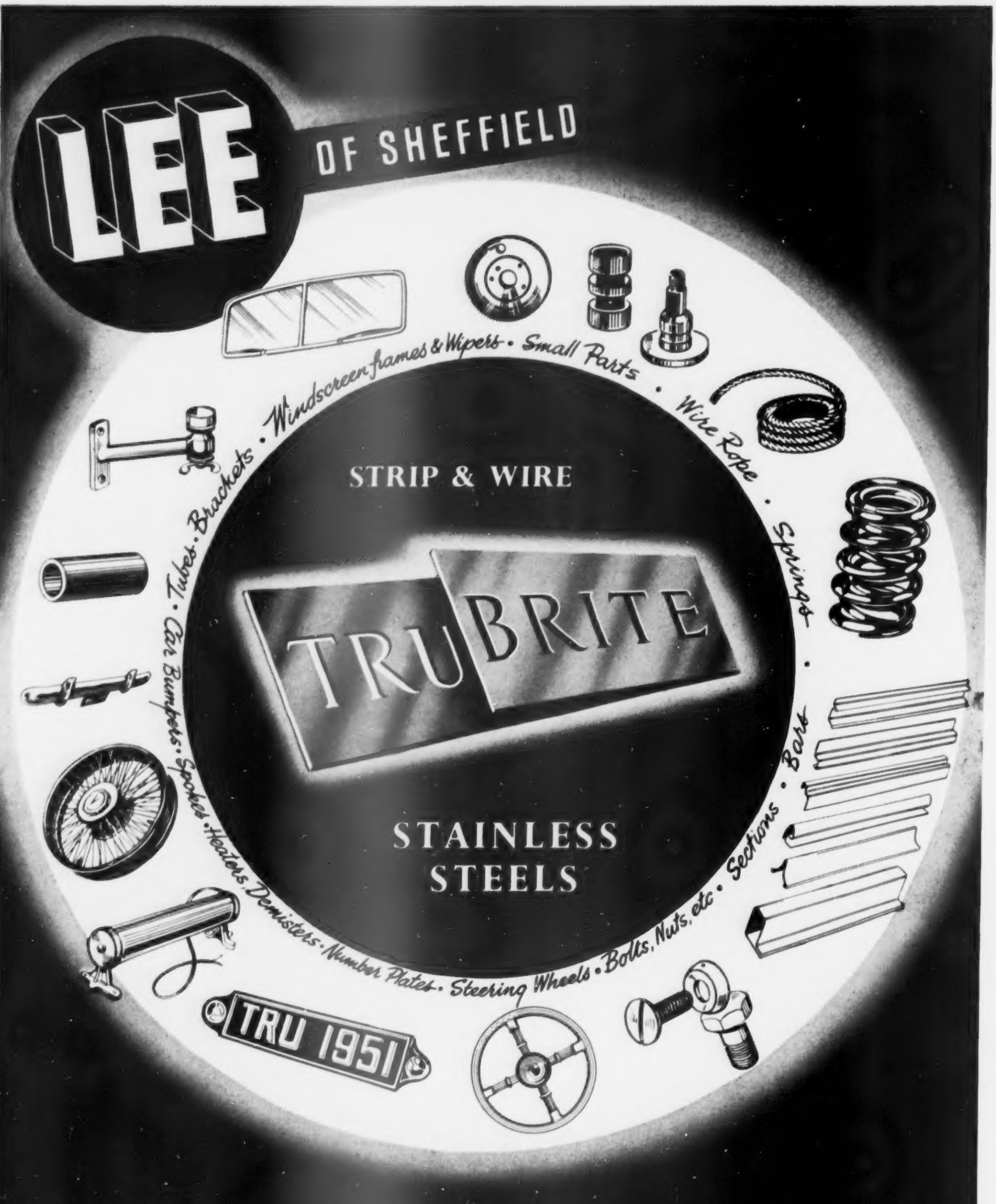
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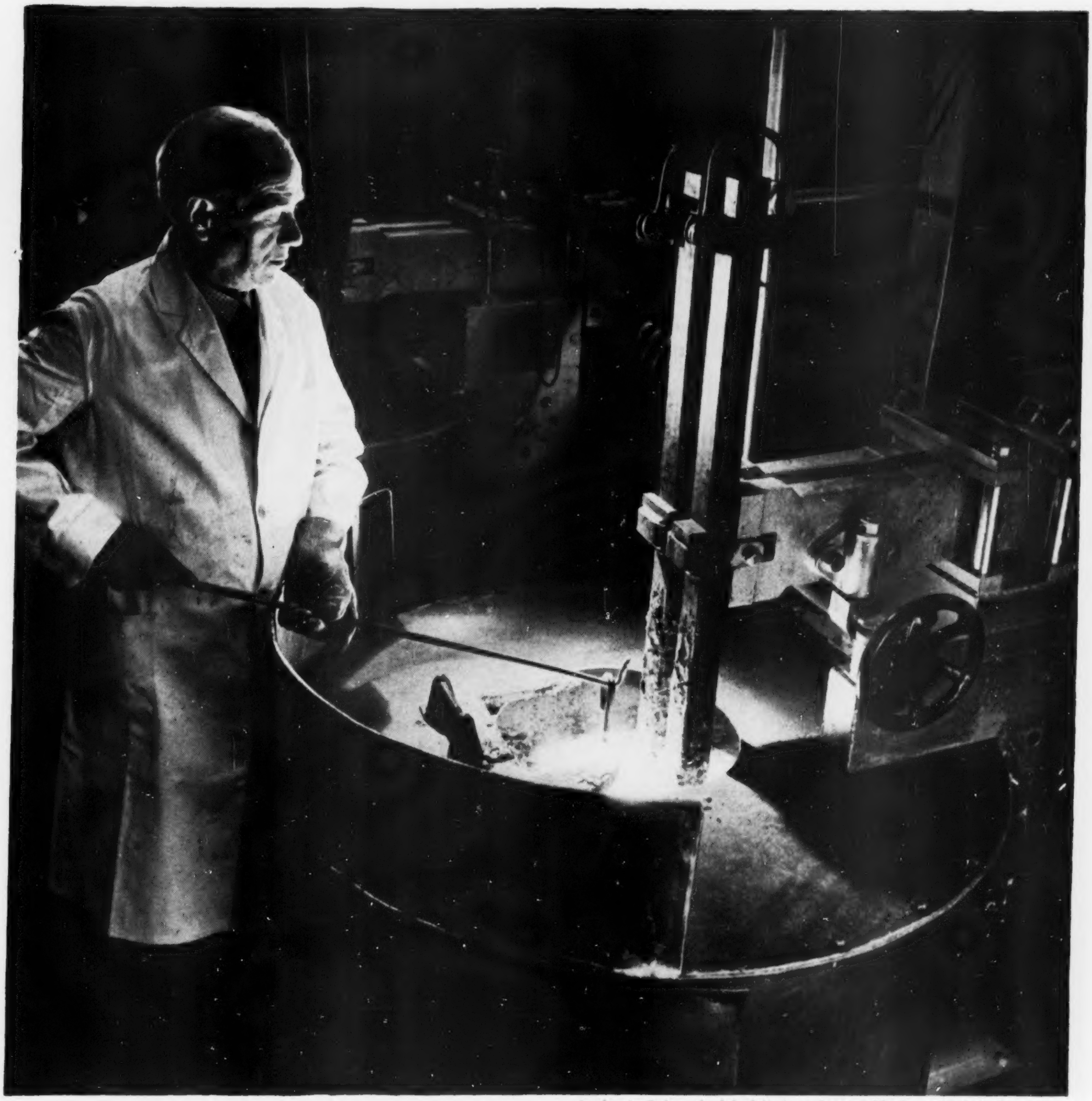

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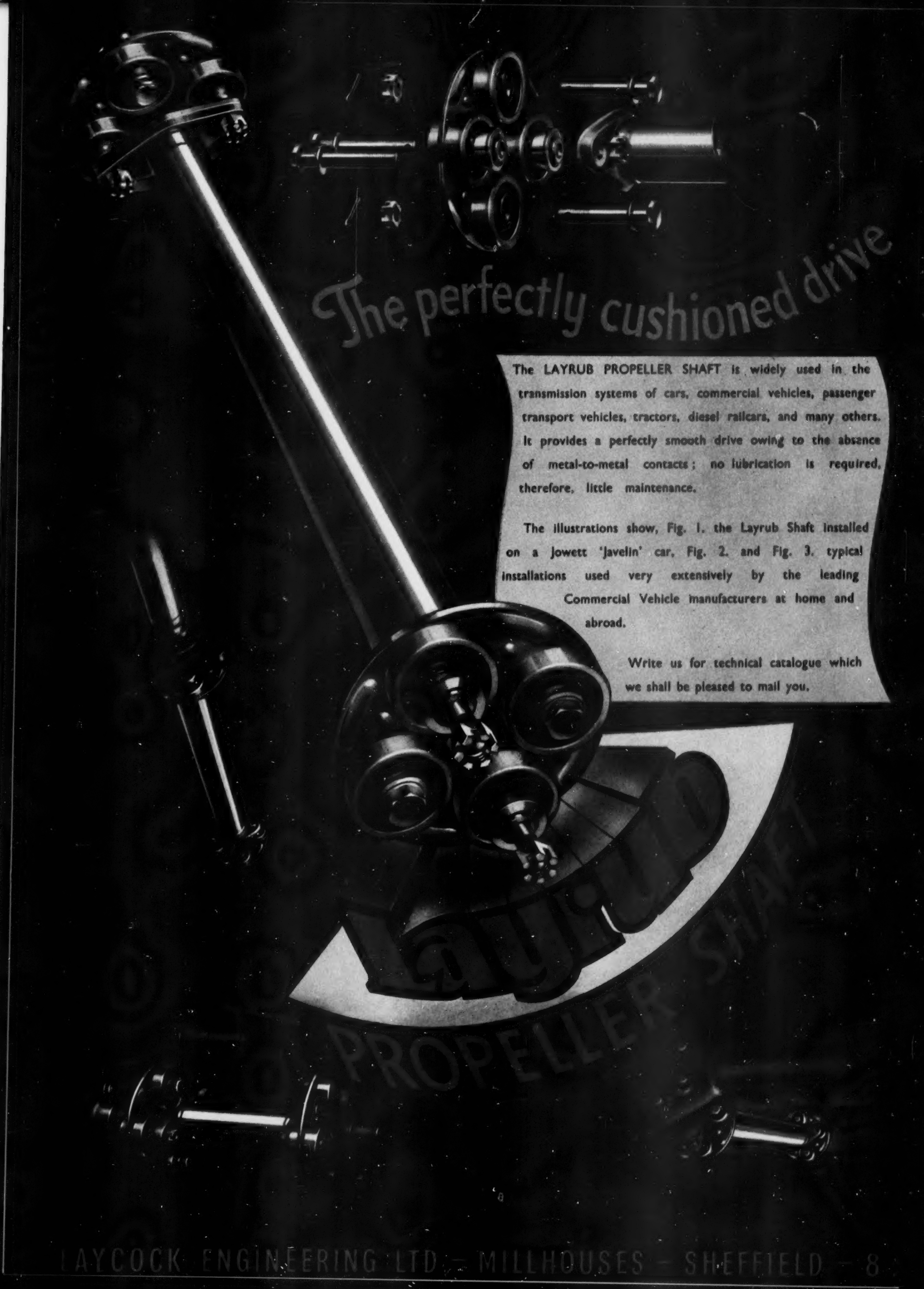
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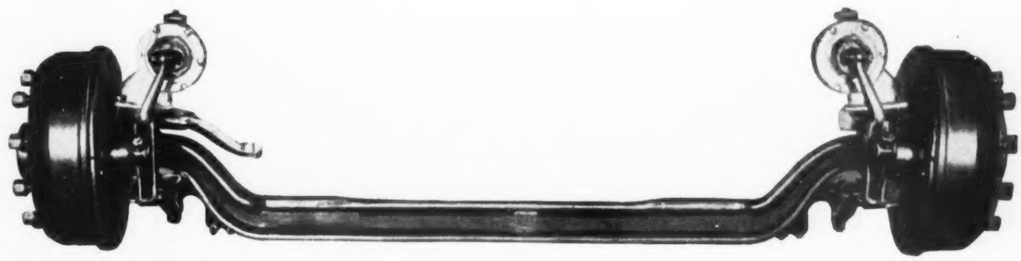

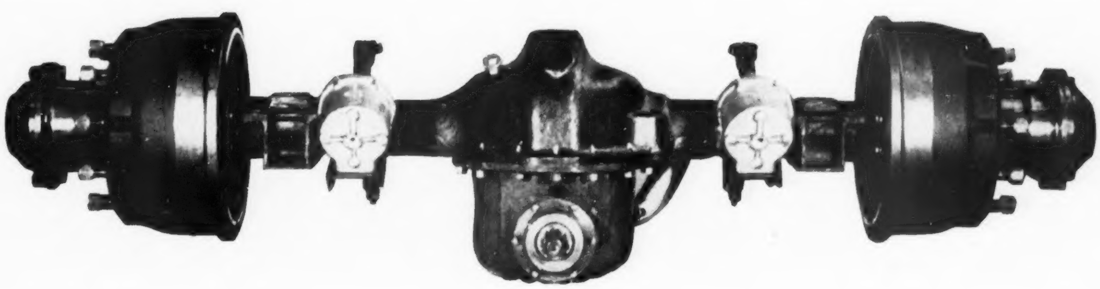

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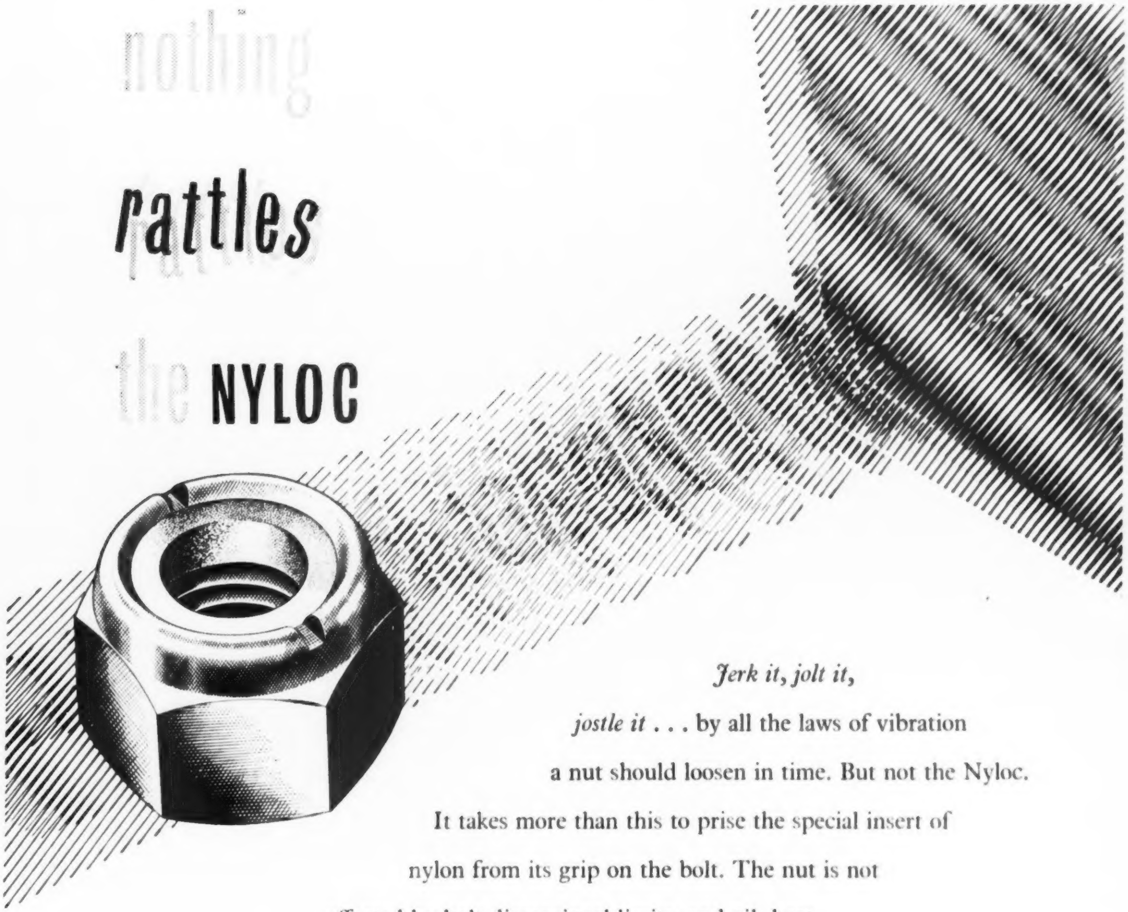

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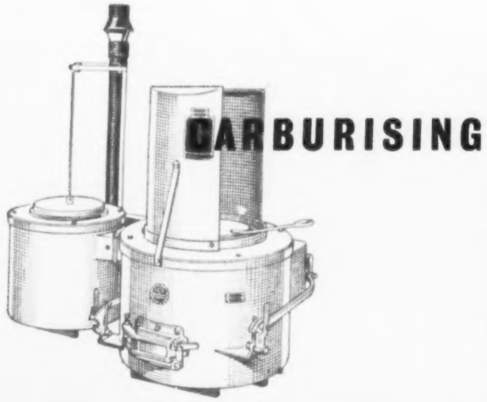

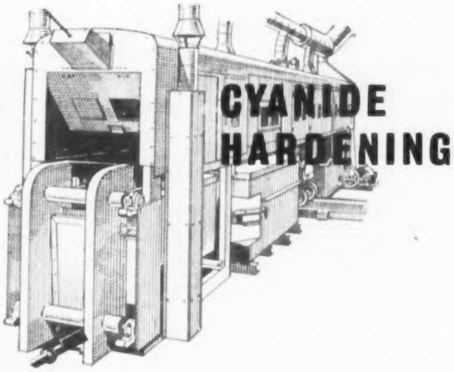

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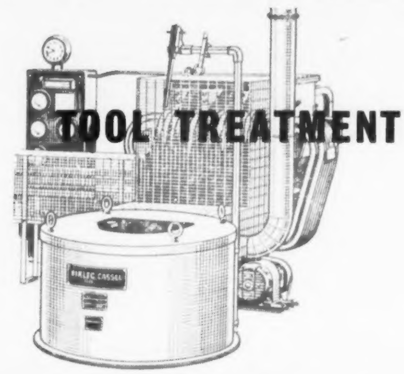

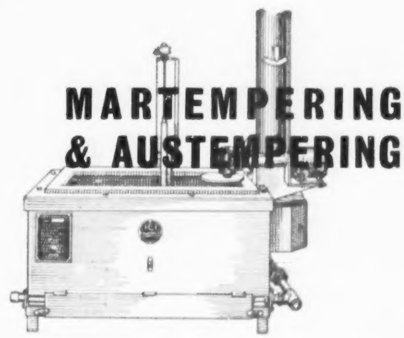

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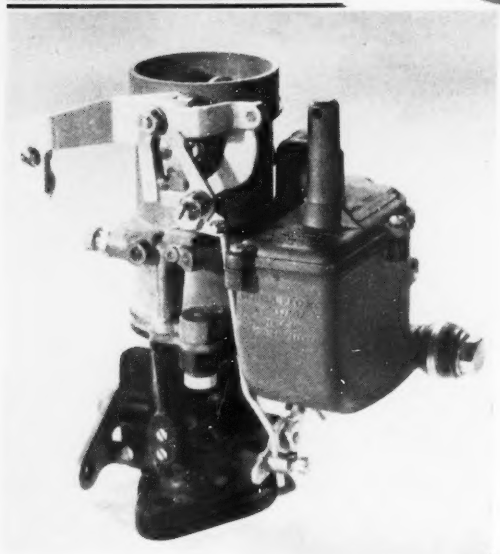

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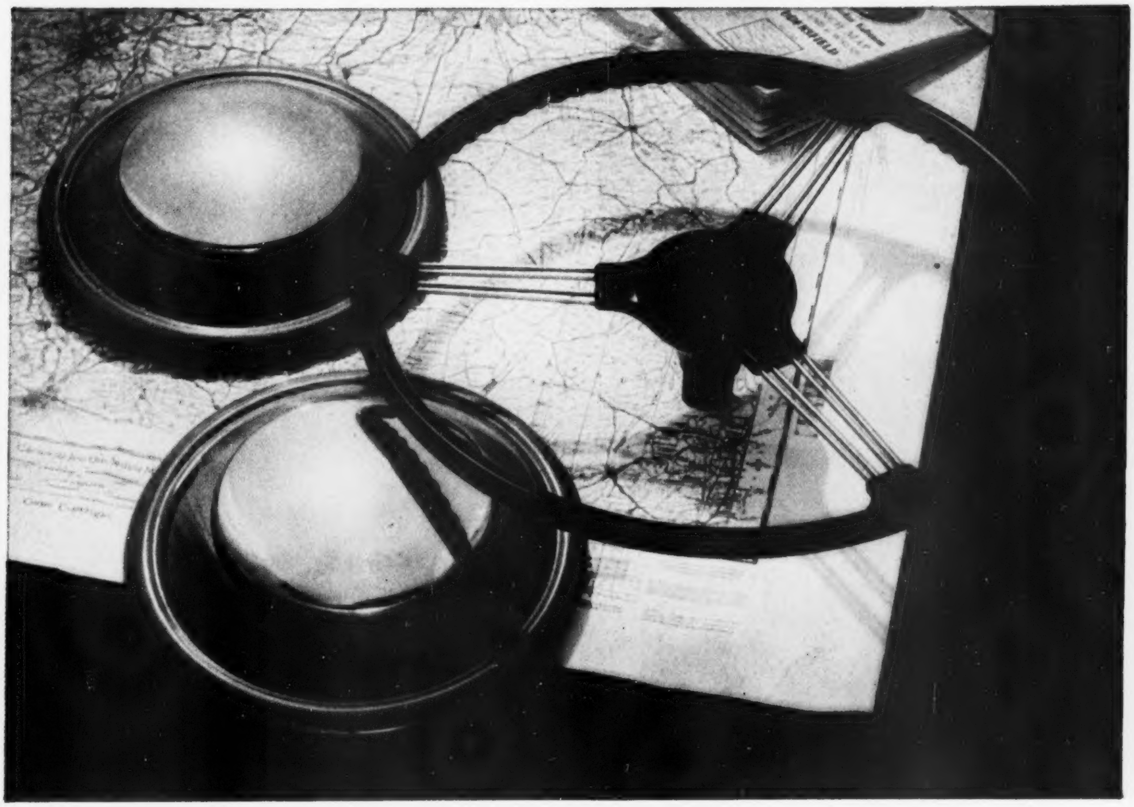


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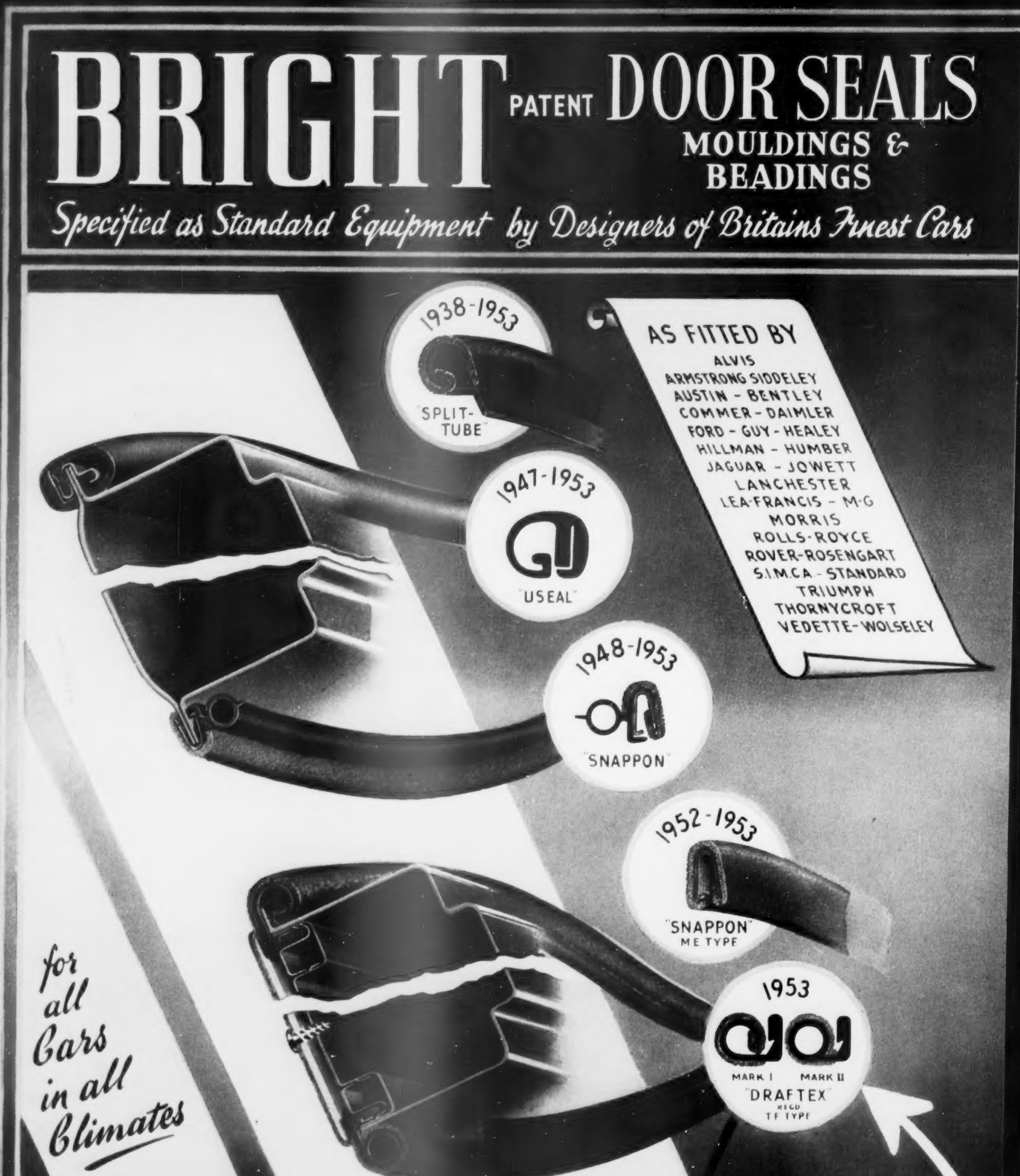
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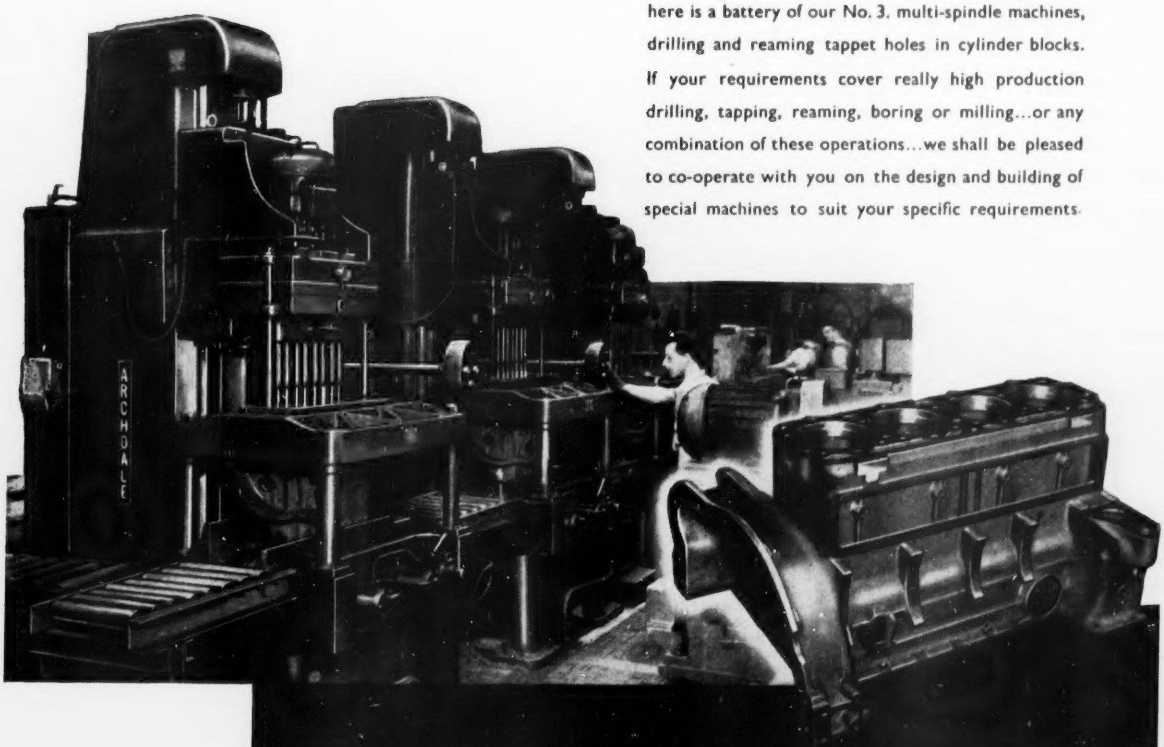

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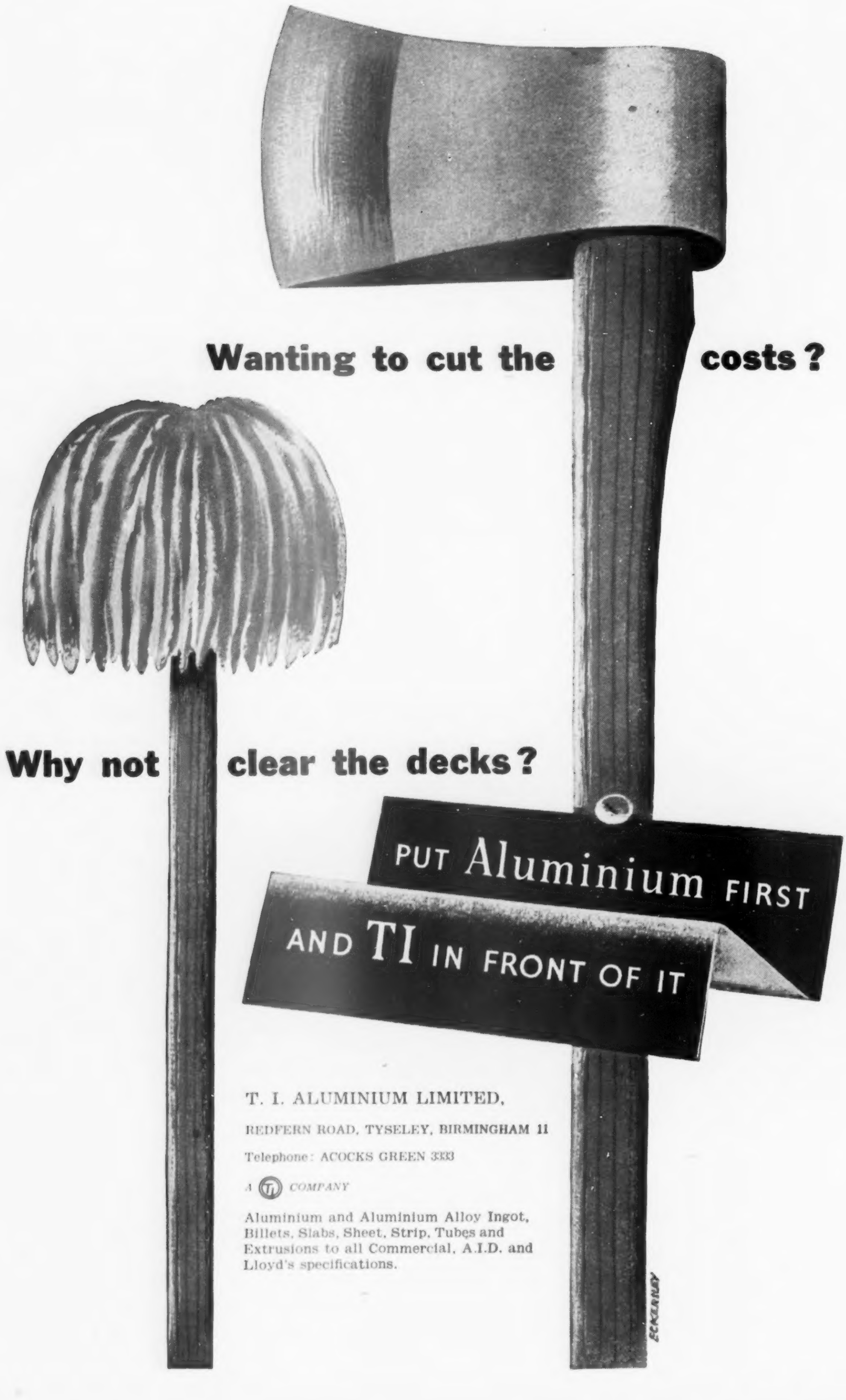

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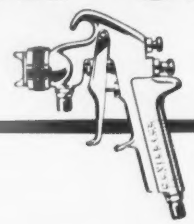


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Design, Materials, Production Methods, and Works Equipment

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Races and Trials

IN the very early days, motor races, hill climbs, and competitions arose from, and were largely dominated by, the spirit of adventure. Such events were "sport"; fairly pure, and comparatively simple. Special purpose cars did not exist, and it was all "good clean fun". Certain constructors soon realized that it was also good business, and some made particular efforts to cash in on this aspect. Even so, it was then a far cry, for example, to the highly specialized design and organization such as lay behind the great German racing *équipes* during the pre-war years.

By that time it was fairly evident that not only was success in competition work good business, but that a valuable façade of National prestige had become associated with prominence in racing and trials. Most of the motor manufacturing nations had entries in the big international events and much of the experience thus gained was reflected in improvement of the ordinary motor car of commerce. Competition work helped to give the world a better product.

Commercial considerations

Whether or no the completely special racing machine justifies itself commercially is a large and debatable subject. Today the sums that can be involved in such ventures are so vast as to be difficult to justify as an individual venture, even granted the enhancement of National prestige and that of an industry as a whole. Where however a more or less standard product is involved, the expenditure is relatively modest in proportion to results that can be gained.

Certain firms have proved quite conclusively that in such cases competition work pays large dividends. Nevertheless, this path once embarked upon, carries definite and serious obligations. The reputation and publicity so gained has to be maintained. Calamitous and "inexplicable" failures must at all costs be avoided. Because of the reputation that has been created, a good showing must at all times be made.

Success cannot of course always be assured. A good and consistent performance can however be guaranteed, providing there is sufficient care and determination behind the effort. This above all implies meticulous attention to detail work, both theoretically and practically. Organization, simple, direct, and of the right kind must obviously be provided, but above all, the technical work must be impeccable.

That this particularly, has not always been of the highest order, past events amply demonstrate. Races have been lost, and nearly lost, owing to some trifling weakness that would never have passed a competent engineer. Humility, not jollity should be the keynote and the guiding spirit in the competition section. Where such work is not sectionalized, success will be largely fortuitous and cannot be effectively planned. However sound and suitable the standard product used in competitions, certain modifications are inevitable, and these are obviously the weak links.

Basic requirements

The fundamental law is that no departure from standard, no innovation, however trifling, can be permitted to go into a competing car without a really gruelling test, surpassing or at least equalling, anything likely to be encountered in the event. To be really on the safe side, even well proved standard units such as gearboxes, axles, shock absorbers, steerings, suspensions, etc. should be fully dismantled and rebuilt under the eye of a really experienced, qualified, and highly skilled automobile engineer, and under view room conditions. Control of a competition section is emphatically not work for a racing expert, but for a top grade engineer who is going to see to it that his cars will at least complete the course.

The casual and haphazard procedure still in some cases followed, can lead only to disaster sooner or later. While the luck holds, the happy-go-lucky spirit will enable an entrant with a basically sound product to "get by" for a time. Sooner or later however, the shock arrives. When the failure or failures occur, the post-mortem invariably discloses nothing mysterious, but just something that could have been foreseen by any sound engineer really giving his mind to the job beforehand. Very seldom indeed does some completely unforseeable and mysterious breakage occur, particularly on every car in the team.

Late modifications

Last minute modifications are an open invitation to the demons of ill-luck. Better withdraw the car or cars from the race than offer such hostage to fortune. The vital point is that in the limelight of international competition, entrants must take more care in their preparations. They must be really serious, perhaps a little sad, but above all, be safe. Why fritter away the exceptional advantages of a well developed and outstanding product, along with laurels already won ?

Given a good and suitable standard chassis, a small,

independent, but high priority competition section is a very worthwhile adjunct to any factory. It must however be staffed by the most highly qualified engineers that money can buy, so that the just penalty for ignominous failure is never likely to be invoked, and creditable "all finished" results at least, can be definitely assured.

Chromium Plate

THERE are still certain features in modern cars which are the subject of much criticism. Perhaps the greatest number and possibly the most justifiable complaints are made against chromium plated components. Nor do these criticisms concern only British cars; the trouble is encountered in vehicles produced all over the world.

That the defects to be found in chromium plate do not all arise from one cause is demonstrated by the large variety of symptoms of failure that may be seen. The troubles include tarnishing which gives rise to a matt surface, rust spots, blisters, flaking and peeling, and corrosion of the base metal under the plate. It is obvious, therefore, that the cure for these troubles is hardly likely to be found by a superficial examination of the problem. Nor is there likely to be one remedy for all the defects.

It is generally recognized that in quality pre-war plate was better than much of more recent production. This has given rise to a mistaken belief that modern plating is thinner than that produced earlier. However, the trouble is of a more fundamental nature. The protection of the plated surface is derived mainly from the nickel, and the chromium is deposited on top merely to prevent tarnishing. It is obvious, therefore, that the root of the trouble lies in the nickel. There is one important change that has taken place of recent years in plating technique. It is the development of the process known as bright nickel plating.

Research approach

It is not suggested that manufacturers should revert to the old methods. Instead, there are two lines of approach along which progress may be made. The first is to make a careful study of the plating obtained by the different processes to determine the reasons for any inadequacy of protection, and the second is to develop satisfactory methods of inspection of plated articles.

Plating is still an art rather than a science, and there is a great need for more exact knowledge of the characteristics of plate, and of the way in which these characteristics are developed during the plating process. When this information is available to all concerned, better methods of control will no doubt be generally adopted. Research on plating problems calls for a well qualified and experienced staff with a highly specialized knowledge. Moreover, it is more likely to be speedily and successfully completed if suitable equipment is to hand.

Investigation costs

The staff of the British Non-Ferrous Metals Research Association have made a start on the work. They have excellent equipment, and are experienced in this type of investigation, but they are hampered for lack of support. This year, about £7,500 has been allocated to cover every aspect of plating research including production and inspection techniques and corrosion problems. This is not nearly enough if the required results are to be obtained within a reasonable time, and this is a problem for which a quick solution is highly desirable.

Hitherto motor and cycle manufacturers have taken the attitude that since most plated components are bought out, it is the platers who should support the necessary research. This attitude is hardly justified for two reasons. Firstly, the motor manufacturers themselves need to apply reliable inspection methods. The tests currently in general use will detect some defects in plate but there are certain inferior types of plating that will pass these tests. Secondly, the motor vehicle, cycle and accessory trades probably take two thirds of the total amount of nickel used for plating in this country. World markets are very competitive, and poor plating can be the cause of serious loss of goodwill. It is felt, therefore, that these trades should accept at least some of the responsibility for effecting the necessary improvements.

The research departments in the plating firms are hard pressed to keep abreast of current developments generally and must find it difficult to undertake fundamental research. In any case, fundamental research can be better carried out co-operatively than individually. Moreover, most plating organizations are small by comparison with the motor manufacturing concerns and the amount they can contribute to their co-operative research association is correspondingly small. Even very limited support from the motor industry would make an appreciable difference to the speed with which results may be obtained in this important matter.

CONTENTS

PISTON DESIGN

Modern Requirements and Design Practices

IN recent years, engine manufacturers have tended more and more to leave the design of the pistons to the component suppliers. From the point of view of obtaining the greatest possible efficiency and the most satisfactory results, this tendency is a good one, but it has led the engine designers, in some cases, to lose touch with the subject. A background of knowledge concerning all the components used in modern power units is essential if the designer is to perform the function of co-ordinating the work of all the appropriate specialists who supply the components. No one can be more conversant with the conditions under which the engine must work than the design and development staff of the engine manufacturer, and only those with an intimate knowledge of these conditions are in a position to take into consideration all the factors relevant to the component design.

Even readers with a limited knowledge of piston design will be able to see that each section of this article could be the subject of a separate paper. It is obvious, therefore, that only a general outline of the features affecting design is possible here. Nevertheless, such an outline may contain much of the background information needed by the engine designer.

In small and medium petrol engines such as are used in private cars, pistons have changed little over the last 10 or 12 years. Perhaps the most important improvements that have been made relate to reducing noise. This has been done by improving design techniques so that it is now possible to incorporate smaller clearances between the piston and the cylinder walls. These improvements have been effected as a result of careful study of piston temperatures which has made possible the more accurate estimation of expansion effects. Another advantage that has accrued from this work is that bearing areas on the skirt and lands have been increased with a consequent reduction in bearing pressures and an improvement in wear resistance.

Another recent trend is the tendency to delete skirt oil control rings. These apparently have been found unessential and the economical advantages to be gained from their omission have not been overlooked. Moreover, the elimination of the bottom ring reduces the piston drag and also ensures that there will be no restriction of the supply of oil to the skirt. This is particularly beneficial under cold starting conditions when petrol dilution of the oil on the cylinder wall may take place. There has also been a tendency to increase the radial thickness of piston rings to give greater pressures between the ring and the wall. A radial depth of D/26, where D is the bore, is common in this country, and even greater thicknesses are employed in the United States.

There have been marked changes in diesel engine pistons during the last decade. They have in part been occasioned by the steady increase in power output from diesel engines. Maximum firing pressures are now as high as 1,100 to 1,200 lb/in^2, and the increases in pressure have been accompanied by proportionate rises in temperature. As a result, a careful study has had to be made of both the mechanical strength requirements and the temperature flow conditions. Emphasis is now placed on sturdiness of construction rather than light weight.

Materials and surface finish

In general, three types of material are used ; cast iron, Y-alloy and a low expansion aluminium alloy containing 10 to 12 per cent silicon. Cast iron pistons are still used in some two-stroke diesel engines because, under normal running conditions, the inertia force, which is the product of their mass and acceleration, can be made to counter-balance the gas load and thus reduce the total force on the piston during the compression and power strokes. Although the forces during the induction and exhaust strokes are greater because of the heavier pistons, the peaks due to gas pressure are reduced. Serious troubles may occur, however, if the engines are allowed to over-speed. A typical diagram of gas, inertia and nett forces on the piston is given in Fig. 1.

Most manufacturers are now using low expansion alloys of aluminium with silicon in preference to Y-alloy. Because of the lower strength of the low expansion alloy at elevated temperatures, the sections must be thicker and pistons made of this material are somewhat heavier. However, except in very high speed racing engines, this is not important because of the tendency, already noted, of the inertia forces to counter-balance the gas forces. Moreover, the weight increase is not so great as might be expected, because the specific gravity of low expansion alloy is somewhat less than that of Y-alloy. There are two reasons for the adoption of low expansion alloys. One is that they are better materials for die casting and the production process is less expensive. The other reason is that cold clearances can be smaller because of the low rate of expansion. This applies not only to the fit of the piston in the cylinder bore, but also to that of the rings in their grooves and the gudgeon pin in its boss.

Production methods have also been improved with the introduction of low expansion alloys. In many cases a single piece core is employed ; it is inserted into and retracted from the mould by an hydraulic ram. Freezing of the molten metal is carried out more effectively by applying water cooling at the thicker sections such as adjacent to the gudgeon pin bosses. The advantages of die casting are well known. They include reduced cost where large quantities are concerned, accurate control over the weight of the casting, and a cleaner finish, which is particularly desirable on surfaces which are not machined.

Surface finish cannot be gauged by eye, since appearances are deceptive and an apparently highly polished piston may have a greater surface roughness than a dull one. Opinions vary as to what is the best manufacturing treatment. In some quarters, a fine turned finish, having an average surface roughness of 18 to 28 micro-inches is preferred, although it is less pleasing to the eye than a ground and polished surface. The best results are obtained with a diamond tipped tool, at high speed, since this cuts through the silicon particles in the alloy. Softer tools may root them out and burnish them over the cavities. The seating of the rings on the sides of their grooves is most important. This should be comparable with that of a clack valve. If there is any clatter, blow-by occurs and both oil and fuel consumption may be affected adversely. On the gudgeon pin the surface finish should be in the order of 3 micro-inches while in the bores of the boss in the piston, the roughness should be about 8 to 10

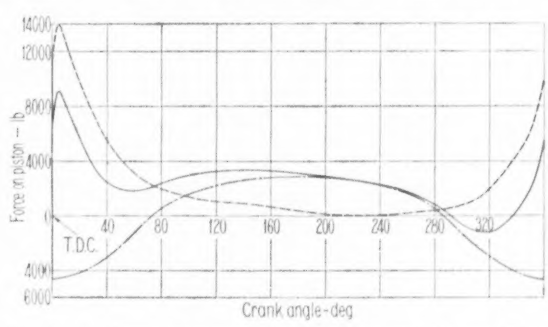


Fig. 1. A diagram showing how the inertia and the gas forces combine to reduce the peaks of the load cycle of a modern high speed diesel engine, — — — — — gas loads, —— · —— · inertia loads, ———— total load

micro-inches.

Tin plating is sometimes applied to the piston to assist running in. The immersion process is usually adopted for this purpose. Another process that is sometimes applied to the surface of pistons is the anodic treatment. The hydroxide coating deposited by this treatment has a hardness value of 700 to 800 Brinell. This hardness cannot be measured by ordinary methods since the depth of the coating is too small, instead it is ascertained by a scratch test. The surface film has a matt finish which retains the lubricating oil. It is particularly beneficial so far as petrol engines are concerned, since it alleviates troubles caused by petrol wash during cold starting. Another advantage of this treatment is that it reduces ring groove wear as long as the hard film remains. When it has worn off, wear proceeds at the normal rate. Anodic treatment is not generally regarded as suitable for diesel engine pistons because the matt surface may key carbon on to the top land.

Wear and seizure

Little wear occurs on the piston skirts. It is rather surprising, considering the soft nature of aluminium pistons, that most of the wear takes place in the cylinder bores and on the rings. However, the explanation for this is that the pressures between the piston rings and the bore are much greater than between the piston and the bore. There is a positive relationship between rate of wear and gas pressures and distance of travel. Consequently, wear tends to be greater in pistons in which there is a large clearance between the rings and the sides of the grooves, since this allows the gas to pass behind the ring and add to the radial pressure. It is impossible to prevent completely the rise of pressure behind the rings, but small side clearances help by tending to restrict fluctuations and to maintain there approximately the average pressure of the four strokes. There are, of course, many more factors that increase rate of wear, but they are not concerned with piston design.

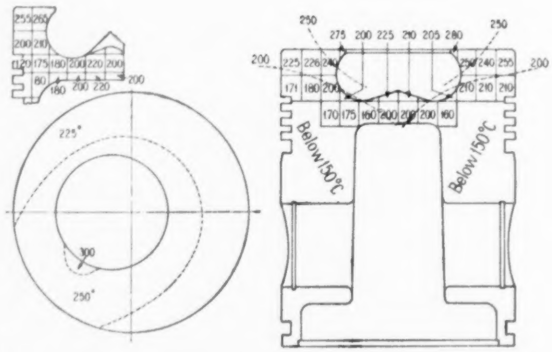


Fig. 3. Temperature distribution on a $4\frac{3}{4}$ in diameter piston for the Sentinel diesel engine. All figures are in deg C

Fairly recently the Dykes ring has been developed, which is arranged so that gas pressure can get behind the ring, but the radial thickness has been reduced so that the spring pressure exerted by the ring is much less. In this way, the pressure between the ring and the cylinder wall is roughly proportional to the gas pressure in the cylinder, so that the effectiveness of the gas seal afforded by the rings is greatest when the need is most. This is done by employing an L-section ring in a stepped groove, the lower portion of the ring, that is the foot of the L-section, is a close fit in the deepest portion of the groove, while the clearance at the top of the ring is greater and gas under pressure may pass into the groove behind it. The outer face of the vertical arm of the L-section bears against the cylinder wall. The foot, because of its greater radial thickness, is stiffer than the upper portion; thus, when the gas pressure is low, the pressure of the lower edge against the wall is greater than the pressure of the upper edge, so that there is a slight scraper action.

During the initial part of the motion of the piston on the firing stroke, inertia forces tend to throw the ring to the top of the groove. Thus, when a conventional rectangular section ring is fitted with large side clearances, the gas pressure escapes from behind the ring so that the radial pressure is reduced over that portion of the stroke during which it is most necessary that it should be maintained. It is claimed that with the Dykes ring, because of the small clearances round the foot of the L-section, the gas pressure between the ring and the base of the upper part of the groove cannot escape, so the radial pressure is relatively constant. So far, this ring has been used mainly in high speed engines, and it is doubtful if it is necessary in lower speed units.

Seizure is a rare occurrence nowadays. However, when it does occur, the piston is more often a victim of circumstances rather than the cause of the trouble. A frequent cause of seizure is the occurrence of excessively high operating temperatures for which the piston has not been designed.

The improvements in design that have led to freedom from seizure have been obtained mainly as a result of careful investigation into temperatures and heat flow phenomena. If the piston profile, is carefully designed and due allowance made for thermal expansion, a large area of the piston skirt and even the lands bear against the cylinder wall. Because of this, bearing pressures are low and the oil film is rarely broken. Fig. 2 is an example of a piston that has been well designed in this respect. It can be seen that the marking spreads out over a large area of the skirt, and above the gudgeon pin it has also been bearing over an appreciable area. Even the top land has been taking its share of the load; this is desirable not only from the point of view of reducing bearing pressures but also to prevent the build up of a hard carbon film which might tend to break away and cause scoring.

Fig. 2. An example of good marking on both thrust faces of a diesel engine piston

Temperatures

Temperature and heat flow control are most important considerations in piston design. The methods of gauging temperatures are worthy of note. In one method, small fuseable plugs which melt at predetermined temperatures may be inserted in the piston, but they are only suitable for measuring surface

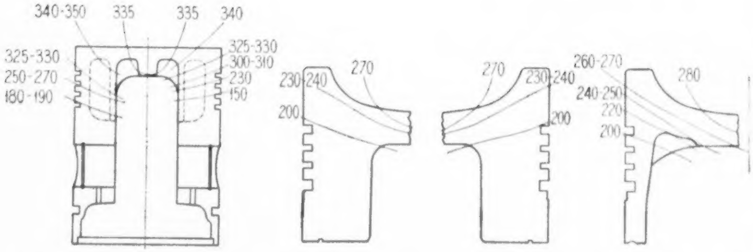


Fig. 4. Left: a temperature survey on a 90 mm diameter piston for the Coventry Victor W.D. 554, and right: the 6 in diameter piston for the National Gas and Oil Engine Company's M4A engine. All figures are in deg C

temperatures. Moreover, the range of melting points of the various eutectics is limited, and there are gaps in it.

A more satisfactory procedure is known as the *recovery hardness* method. For development or experimental work, the engine is run for 50 hours at a constant rating; it is then dismantled and the pistons are split on a vertical plane through the centre of the gudgeon pin bosses. The section is next marked out in small squares and in the centre of each a Brinell hardness test is taken. The hardness is compared with known standards, and gives a reliable indication of the local temperature maintained during the running period. A large diameter ball is used for the hardness tests, since a small one or a diamond point might contact a particle or group of particles of silicon, and thus give a false hardness reading. The 50-hour running period is regarded as the minimum soaking time necessary to ensure that all changes in the grain structure have ample time to occur.

Another method is to split the piston, as before, and etch the surface so that the grain structure may be compared with known standards. With both methods it is usual to split the piston on another vertical plane at right angles to the first one and carry out another temperature survey. In the event of seizure, or any other trouble, it is possible, by using the *recovery hardness* method or by etching, to determine the conditions under which the piston was running prior to failure. Any local over-heating that has occurred as a result of seizure will not affect the results of the temperature checks since they could only have occurred over a relatively short period and changes in the grain structure will not have had time to take effect.

Temperature sensitive paints may also be used. These can be quickly and easily applied, but they have several disadvantages. They give only surface temperatures and the results are not particularly accurate. In addition, when comparative tests are made, care must be taken to ensure that all tests are carried out for the same length of time.

Perhaps one of the best laboratory methods of measuring temperatures is to embed contact thermocouples in the piston, so that at the bottom of each stroke they make contact with spring loaded pads. The principal advantage of this method is that temperature readings may be taken while the engine is running. However, the installation of the thermocouples is not always easy. Moreover, inaccuracies may arise because of the effects of inertia loading and of oil on the elements.

Temperatures vary appreciably with different engine designs, and they are governed by a number of factors. The type of liners employed and their fit in the cylinder block have a marked influence. The most effective cooling is obtained with wet liners, and bores machined in the cylinder block are usually slightly better than press-fit dry liners. In good engine designs, the top of the water jacket is level with or higher than the upper piston ring when it is at the top dead centre position.

The temperature distribution over the piston crown is affected by, among other things, the valve arrangement. Immediately below the exhaust valve the temperature is often about 10 to 15 deg C higher than over the rest of the crown. When pre-combustion or swirl chambers are employed, the

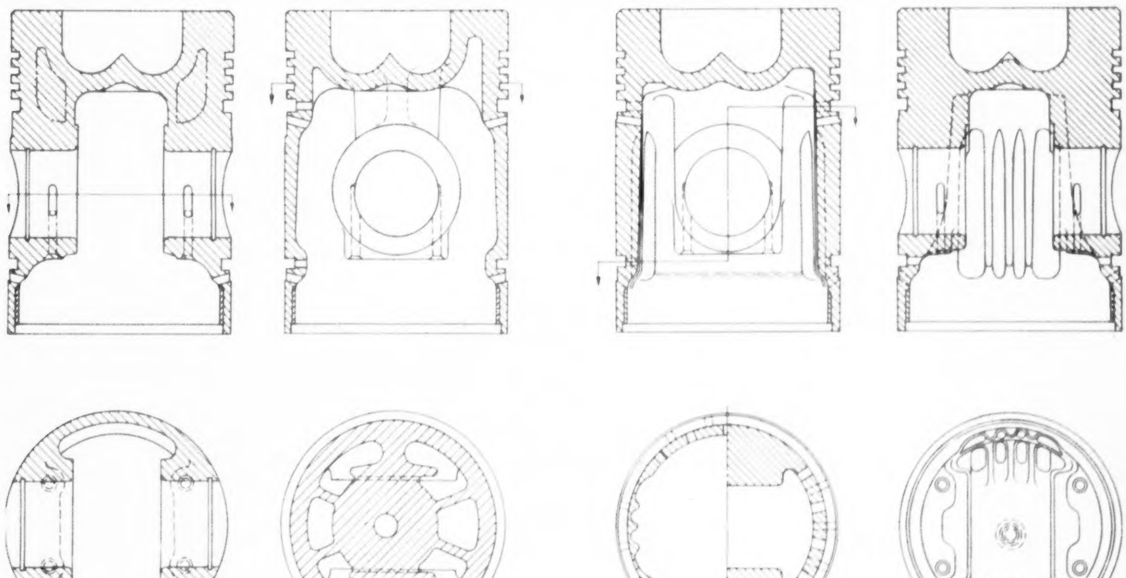


Fig. 5. Two examples of 105 mm diameter pistons. To facilitate heat flow to the skirt, radial webs under the crown have now been superseded by transverse ones, vertical ribs are incorporated on the skirt, and sections around the combustion bowl and ring belt have been thickened

discharge from these chambers can also have a marked effect on temperature distribution. Asymmetry of crown profile will also give an uneven temperature effect. For instance, when the combustion bowl is offset from the centre of the crown, the temperatures between the edge of the bowl and the side of the piston, towards which it is offset, are higher than those on the other side. The ideal to be aimed at is an even temperature distribution over the whole crown in order to avoid distortion, which can be serious if it extends to the ring belt. Temperatures should certainly be kept below 400 deg C since the physical strength of aluminium alloys decreases appreciably above this point. Many authorities consider that the maximum crown temperatures should not exceed 375 deg C for overload rating and 350 deg C for continuous rating. Typical diagrams showing temperature distribution over pistons are given in Figs. 3 and 4.

Engine rating obviously has a marked effect on piston temperatures, but it is surprising how often this is overlooked when manufacturers increase the rating. Some diesel engined vehicle operators depart from the recommended fuel pump settings in an attempt to improve fuel consumption. In some quarters, there is a belief that if the delivery stroke of the injection pump is reduced the resultant loss of power can be offset by advancing the timing, and fuel economy results. However, so far as we know, no reliable figures have been published in support of this theory. Altering the timing in this way has been known to raise piston crown temperature, and to increase wear in the gudgeon pin and bushes because of the excessive temperatures in that region and because of the high peak gas pressures. At full load, the heat flow from the combustion gases to the crown of the piston is the equivalent of about 9 per cent of the b.h.p. in a petrol engine, and approximately 17 per cent in a diesel unit. The higher figure is obtained because of the greater gas density and turbulence.

In all pistons, the temperatures occurring at the ring belt are most important and, in general, they should not exceed about 212 deg C if straight mineral lubricating oils are to be used, otherwise carbon and lacquer formation may be excessive, and ring sticking may occur. When detergents are employed the allowable temperatures may be at least 25 deg C higher. However, higher temperatures are possible with well designed pistons in which deformation due to thermal expansion is reduced to a minimum. If taper sided rings are employed, higher temperatures are not so detrimental since such rings tend to free themselves from sticking. The disadvantages of this type of ring are that blow-by is greater and re-grooving in the field is beyond the capabilities of many operators and service depots.

Several design features have a marked effect on ring belt temperatures. It is important, of course, that the thickness of metal between the base of the ring grooves and the inner wall of the piston is adequate to conduct the heat from the crown to the skirt. In earlier designs this point was often overlooked, and undue restriction of flow was caused by the thin section between the lower ring groove and the inner wall of the piston. Another feature that can be introduced to control ring groove temperatures is a *heat break groove* around the top land immediately above the compression rings. The object of such a groove is, as its name implies, to prevent the direct flow of heat from the crown to the grooves and to deflect it inwards so that it will by-pass them.

In earlier designs, the coring was extended up past the ring belt so that the section round the combustion bowl was more or less of constant thickness. However, it was found that this restricted the heat flow from the centre of the crown to the skirt and directed it too close to the ring belt. In modern designs, therefore, the crown is more solidly made. This is illustrated in Fig. 5 which also shows how thicker sections are maintained below the lower ring groove and how vertical ribbing is employed to transmit the heat down the length of the skirt. Thickening of the crown sections has been known to lower crown temperatures by as much as 30 deg C while raising skirt temperatures by about 15-20 deg C. An incidental advantage of the vertical ribs is that they help to stiffen the skirt, but this is not the primary reason for their incorporation.

There are two reasons for directing so much effort towards arranging for the heat to flow down into the skirt. The most important is that by far the greater proportion of the heat absorbed by the piston is conducted away through the skirt to the cylinder wall. This is because the area of contact between the skirt and the cylinder is much greater than between the lands and the wall. In many quarters it is considered that there is little heat transfer through the piston rings. The reason for this is that oil and lacquer films form a fairly effective insulation, and it is thought that the rings are more or less floating at all times and have no very effective metal to metal contact with any of their surrounding components. Moreover, the thermal conductivity of aluminium is about three times that of iron.

It should be mentioned here that some investigators hold that most of the heat transfer to the cylinder walls takes place through the rings. This opinion is based mainly on measurements of temperature gradients between the crown and skirt of different pistons. Some of these measurements have indicated that the steepest gradient, or the greatest fall in temperature, takes place over the ring belt, and that there is only a small temperature difference between the piston skirt and cylinder wall. A further argument advanced

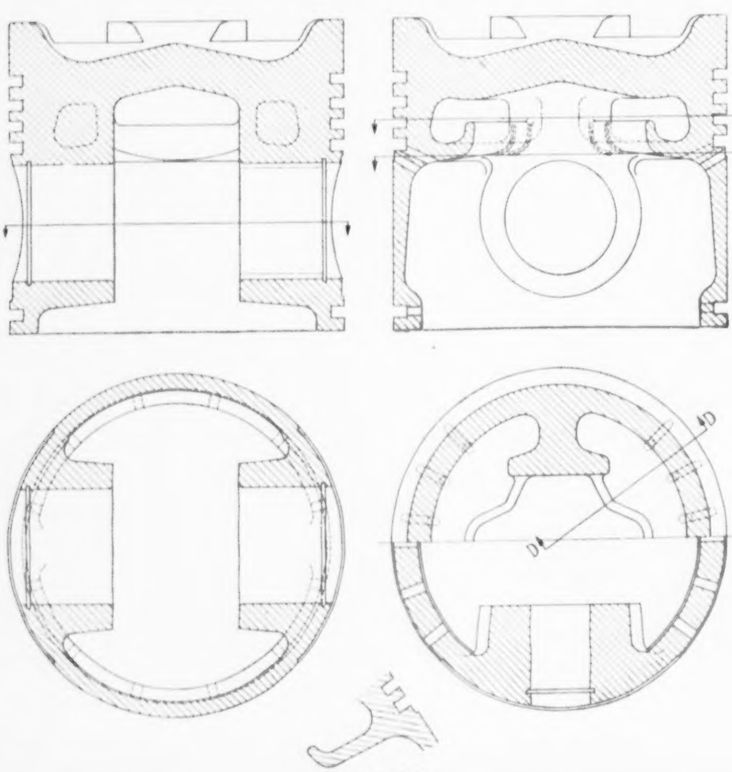


Fig. 6. A 7 in diameter, Ricardo Cocktail Shaker type oil cooled piston.

to support this theory is that the pressures between the rings and the cylinder walls and ring grooves are much greater than those between the pistons and walls. It is also claimed that active oil films are very effective convectors of heat.

However, we do not know with what pistons these results were obtained, and they may have been due to the existence of a thin section below the top scraper ring, giving rise to a heat brake effect. This would prevent the flow of heat into the skirt. Even under these conditions, it seems that the greater proportion of the heat transfer may have taken place between the lands and the walls rather than through the rings.

Heat losses in other ways from the piston are not very great. Radiation is small because of the fairly high temperatures of the gas inside the crankcase. Convection is not as effective as might be imagined, since the gas pocketed in the piston tends to move up and down with it, so there is very little flow which might cause cooling.

In some large bore diesel engines where the centre of the crown is remote from the skirt, it has been necessary to introduce oil cooling. Two designs of oil cooled pistons are shown in Figs. 6 and 7. The simplest is the Ricardo Cocktail Shaker. In this design, a cavity is incorporated beneath the crown, and extends round the inner wall of the piston between the gudgeon pin bosses. Oil is directed up into the cavity by a nozzle integral with the top end of the connecting rod. The jet strikes the underside of the crown of the piston and is deflected outwards into the cavity and, as the piston moves up and down, the oil is thrown about violently between the base of the cavity and the underside of the piston crown. It then flows away through the opening immediately above the gudgeon pin and small end assembly to the crankcase. The heat taken from the piston is either disposed of by a heat exchanger in large engines, or in small ones by crankcase radiation.

The second oil cooled piston illustrated is more complex. In this design, a chamber is cored immediately below the crown. In the base of this chamber is a circular hole, spigoted into which is a plunger guide. The guide is flanged to take the set bolts securing it to the base of the chamber. It carries a hollow cylindrical plunger, the base of which is shaped to seat saddle fashion on the round upper end of the connecting rod. Contact between the two components is ensured at all times by a compression coil spring between the saddle and the plunger guide. The centre of the lower face of the plunger is milled out so as to form a chamber of such a width that an angle subtended between its edges and the centre of the gudgeon pin is greater than the angular motion of the connecting rod. This means that an oil hole drilled vertically in the top of the rod is at all times in communication with this chamber.

Cooling oil is passed from the big end, through a vertical drilling to an annular groove around the small end bush and its housing. Thence it is passed into the chamber in the saddle and up the hollow barrel of the plunger. It then impinges on the centre of the piston crown and is deflected outwards between a flange on the top end of the plunger housing and a circular rib round the centre of the crown. Its velocity through this restriction is increased sufficiently for it to circulate round the rest of the chamber beneath the crown, whence it returns, through holes drilled through the spigot portion of the plunger guide, and drains away to the crankcase. In most oil cooling arrangements, a non return valve must be incorporated in the connecting rod big end, or the crankpin must be designed as a rotary valve to prevent reverse flow of oil due to inertia effects.

These may appear to be rather complicated arrangements for effecting oil cooling. However, it has been found that the simpler method of directing a jet of oil up on to the piston crown is not so effective. This is because the oil is not in contact with the piston long enough to effect adequate heat transfer. It can be made more effective by using a larger jet, but this introduces problems both in regard to oil supply and oil control.

Profiles

Piston profiles are mainly determined by temperatures. Above the ring belt, for instance, where the temperature is highest when the engine is at its normal running temperature, the overall diameter must be smallest. If the piston is well designed, the expansion will take up the clearance until the optimum running fit is obtained in the bore. Piston profiles are important for two reasons. Firstly, the bearing area between the piston and cylinder must be as large as possible. The smaller the area, the higher is the bearing pressure and therefore the greater the tendency to scuffing. The second reason is that unless the piston section is reasonably circular at its operating temperature, oil control is difficult.

Over the past two or three years, considerable progress has been made with regard to design procedures for determining accurately the overall dimensions of pistons. It is only during this period that principles have been established which enable manufacturers to produce direct from the drawing board prototype pistons which, when fitted to the engines and run, exhibit the correct bearing characteristics. Thus, little or no development work is required. It is the establishment of these principles which has led to pistons being designed in such a manner that the land above the ring belt bears on the cylinder wall. Hitherto, it was the practice to design in such a way that there was a large clearance between this part of the piston and the cylinder walls. In modern diesel engine pistons of low expansion alloy the maximum cold clearance is no more than 0·004 in per in diameter at the top land.

The earliest attempts to allow for differential rates of expansion between

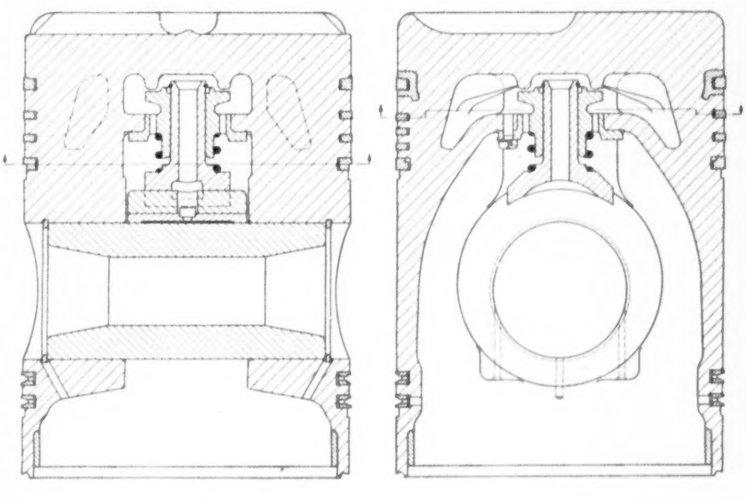


Fig. 7. An $8\frac{1}{2}$ in diameter, Specialloid pressure fed oil cooled piston. Alternative ring arrangements are shown in each elevation.

the crown and skirt led to the adoption of a single taper profile. In this type of piston, the largest diameter was, of course, at the base of the skirt and the smallest at the crown. The next stage in the design progress was to incorporate two tapers, one from the skirt to the ring belt and the second from the ring belt to the crown. Yet a further refinement was introduced when three tapers were adopted, one over the length of the skirt and the other two over the lands.

All these designs were incorporated in conjunction with an oval plan-form. In the later designs the ovality was varied, again to allow for differential rates of expansion over the length of the piston. The need for ovality arises, of course, because of the relatively large mass of metal around the gudgeon pin bosses. This thick section naturally expands more than the thin ones in the other parts of the skirt. Fig. 8 shows a typical diesel engine piston profile diagram.

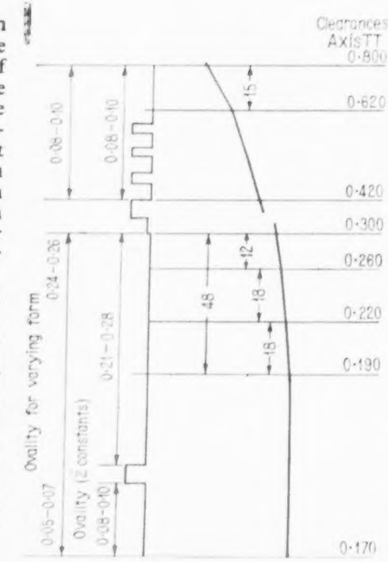


Fig. 8. A diagram showing typical diesel engine piston profile dimensions. All figures are in mm, the piston diameter is approximately 125 mm, and clearances are measured in a vertical plane perpendicular to the axis of the gudgeon pin

Piston types and constructions

There are three main types of piston. The first is the split skirt, Fig. 9, the second is the T-slotted skirt, Fig. 10, and the third the solid skirt piston, shown in Fig. 11. Diesel engine pistons, because of their high loading, are almost invariably of the solid skirt type. The other two forms of piston are used in some petrol engines because less accurate machining of the cylinder bores is called for, and the natural resilience of this type of piston enables it to take up most of the clearance between the skirt and the cylinder wall, and results in silent running both at normal operating temperatures and when cold. A disadvantage of these types of piston is that the drag is greater. The full split skirt is favoured by servicing departments because it will adapt itself to worn cylinder bores and alignment is not critical.

Most slotted or split skirt pistons incorporate a thermal slot beneath the ring belt. Thermal slots are horizontal and are joined usually at the centre by the vertical slot, or split in the skirt. These slots are intended not so much to prevent heat from flowing down to the skirt as to control the flow and obtain an even temperature distribution. Their length and position is decided by temperature checks during the development stage. In some cases, a thermal slot only is incorporated.

Not only is it necessary to determine with care the profile of the piston to obtain good bearing on the cylinder walls, but also the structural design must be such that gas, inertia and bearing loads are adequately catered for. Piston crown support webs have been changed of recent years. Earlier models has radially disposed webs, but more recently it has been found better to incorporate webs positioned transversely relative to the axis of the gudgeon pin. There are two reasons for this. One is that webs arranged in this way help to transfer heat down to the skirt; they are not needed of course where there is a metal path from the crown through the gudgeon pin boss support struts to the skirt. The other reason is that the ring belt adjacent to the thrust faces of the piston would not otherwise be adequately supported, as is the case above each piston boss where it is strengthened by the struts. Without these supports the tendency is for the ring belt to deform telescopically. This tends to close the ring clearance partly and contributes to ring sticking.

Piston bosses and their support

Support for the bosses is an important requirement. Some manufacturers maintain that the bosses should not overhang the inner faces of the support struts which are incorporated between them and the crown, while others incorporate this overhang and drill oil holes vertically through the overhung portion of the boss so that lubricant may fall into them from above. Under working conditions there undoubtedly must be some deflection of the gudgeon pin which, because of its alternating characteristics, will apply to the overhung bosses bending loads of a nature liable to cause fatigue failure, particularly at the junction between the boss and the lower end of the inner face of the strut. These bending loads will, of course, be superimposed on the compressive and tensile ones that must in any case be carried by the strut as the piston moves up and down. It is regarded as a sound general engineering principle to avoid wherever possible offsets liable to cause such conditions.

In some of the earlier designs without overhang, the strut was of I-section, one flange being above the inner face of the boss and the other being formed by the ring belt. More recently, a solid rectangular section has been used. It is claimed that this section leads to better heat flow conditions, but it is also most convenient when hydraulically operated cores are used in production. Care must be taken to ensure that the temperature adjacent to the gudgeon pin bearing surface does not exceed about 260 deg C, other-

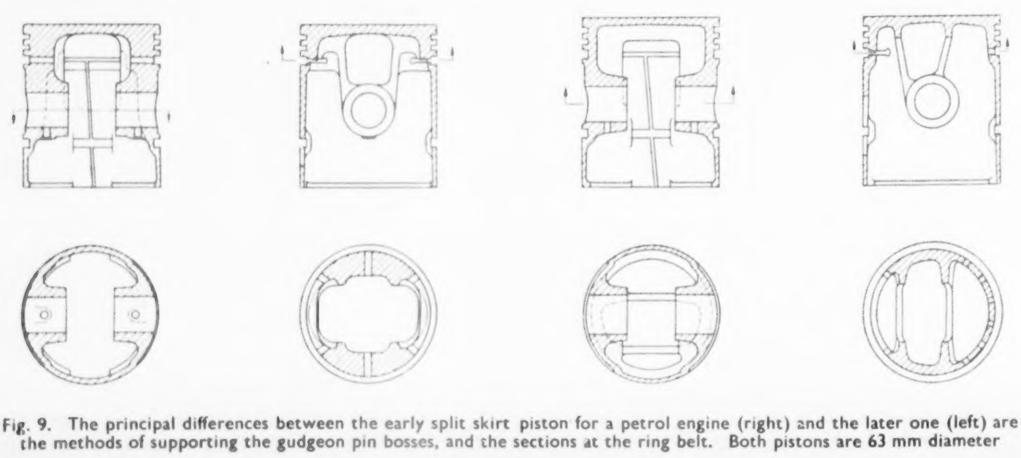


Fig. 9. The principal differences between the early split skirt piston for a petrol engine (right) and the later one (left) are the methods of supporting the gudgeon pin bosses, and the sections at the ring belt. Both pistons are 63 mm diameter

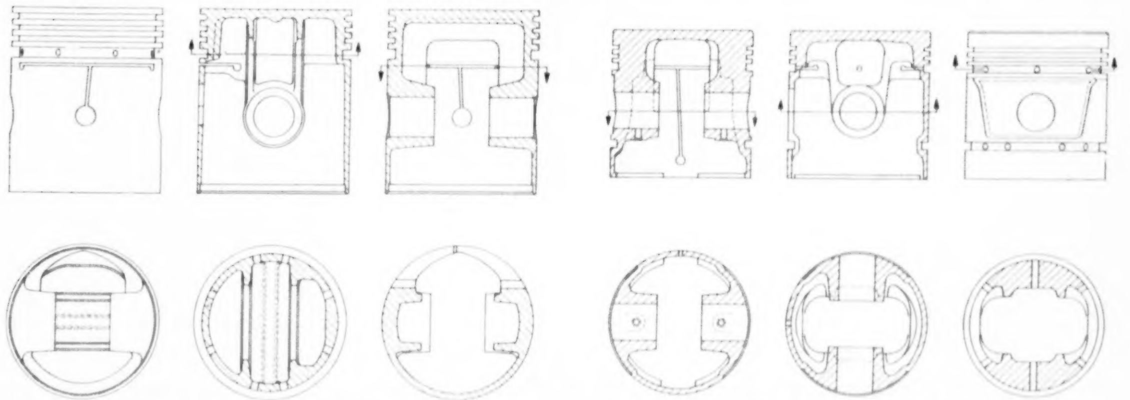


Fig. 10. Specialloid slotted pistons for the 79·37 mm bore Willys Overland (left) and the 73·5 mm. bore Morris Commercial petrol engine (right)

wise collapse may take place under fatigue loading.

Struts of V- or W-shape have also been used above piston bosses. However, some designers do not consider that these are very satisfactory because under the elevated temperature conditions at the crown, they tend to spread under load and cause deformation of the ring belt. This spread is often a plastic rather than an elastic phenomenon and the deformation is permanent.

Another practice that has been adopted is that of slightly tapering the inner ends of the bores of the bosses to allow for the bending deflection of the gudgeon pin. Although the taper relieves the load on this otherwise heavily stressed portion of the boss, it would appear to have the disadvantage of reducing the fully effective bearing areas. Where solid vertical struts are employed, lubrication is often effected by splash passing up through two vertical holes drilled from underneath the boss in such a manner that they break into the bore tangentially. With this arrangement, the oil passes into the bore at points where it is lightly loaded. In higher rated diesel engines, lubrication is sometimes effected through holes drilled downwards to the boss from the relief below the top oil control ring. Various combinations of one or two holes from above, with or without one or two holes from below may be used to suit the design of the piston.

An approximation to the bore size required in the gudgeon pin boss may be obtained by basing the design on a maximum vertical loading of 5,680 lb/in² on the projected area of the bearing. In calculating the size of gudgeon pin required it is satisfactory to design for a bending deflection of 0·001 in at the centre, and for 0·001 in flattening of the hollow pins. The dimensions required to give these deflections may be calculated from the conventional bending theory, applying the estimated gas and inertia loads. Fully floating gudgeon pins should be a push fit in the boss at room temperatures, and at 75 deg C should be free. This ensures that under starting conditions, when lubrication is hardly adequate, all movement takes place between the small end bush and the pin. Gudgeon pins that are clamped in the connecting rod small ends should be free to rotate in their bosses at room temperature.

In two-stroke engines the gudgeon pin loads are very critical, because gas loads are more continuous and temperatures are higher. It is advisable to bush the piston bosses in two-stroke engine pistons, and great care is necessary in providing for lubrication. In some designs the bushes are cast in, and in others they are pressed in. Illustrated in Fig 12 is a small two-stroke petrol engine piston. The bushes are turned from bronze hexagonal bar, and hexagonal flanges are left at their outer ends. They are cast in the boss and the flanges ensure that there is no possibility of their rotating under load.

The advantage of casting in the bushes is that at elevated temperatures they are less likely to come loose as a result of the differential rate of expansion between phosphor bronze and aluminium. However, it is necessary to use bronze alloy, the properties of which are not adversely affected by the high temperatures experienced during the casting process. The phosphor bronze alloy normally used for bearing bushes would anneal at these temperatures. In two-stroke diesel engines, it is often necessary to pressure lubricate these bushes. One way of doing this is to incorporate blanking plates in each end of the gudgeon pin, and to feed oil from the big end to the small end and thence into the hollow pins and then through radial drillings to the bushes.

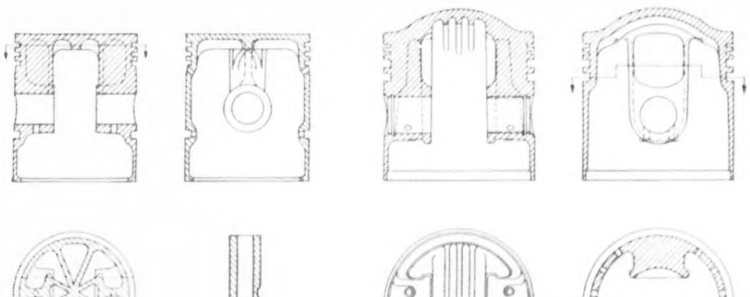


Fig. 11. Left: this early type, 63 mm diameter, solid skirt piston for a petrol engine has radial ribs under the crown, and I-section boss support struts. Right: the 78 mm diameter solid skirt piston for the Lagonda engine has a recess machined in each boss support strut for lightness, and the inner edges of the boss and strut are lipped for strength

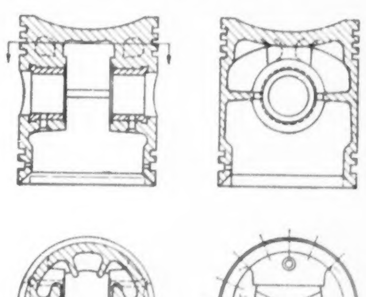


Fig. 12. Cast-in bushes are employed in this 56 mm diameter piston for a two-stroke engine. Hexagonal flanges on the bushes give axial and rotational location

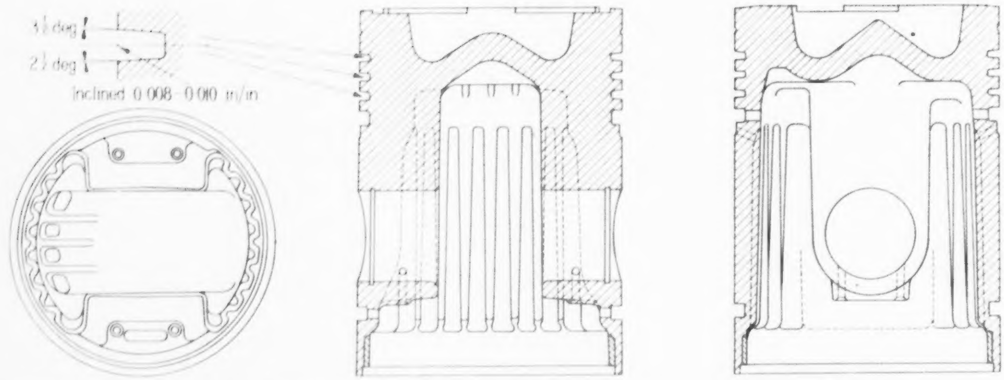


Fig. 13. This 4·8 in diameter piston for the Leyland six cylinder oil engine has an open variable concave combustion bowl and taper sided and uptilted compression rings

The ring pack

So many theories and opinions have been expressed concerning piston rings and so many varieties are available that a whole volume could be written on the subject. It is, therefore, impossible to give in such a small space as is available here more than a few remarks of general interest. Almost every engine has different requirements with regard to ring pack arrangement, particularly so far as oil control is concerned. Moreover, the trend towards the use of low viscosity oils has further complicated the problem.

With low viscosity lubricants, the fundamental requirement is to maintain proper oil control without using high radial pressure rings which, because of the greater friction, would offset the advantages sought by using low viscosity oils. One arrangement that is recommended for a 120 mm diameter piston is as follows. At the lower end of the skirt, a ¼ in wide slotted scraper ring with a radiused upper edge is fitted. Because of its tendency to ride over the oil film on the upward stroke it is possible to use a lower radial pressure with this type of ring, and its radial thickness is 0·177–0·169 in. The radial thickness of all the other rings is 0·190–0·182 in.

The oil control ring above the gudgeon pin is stepped externally and also has a radiused top edge. Its face and step widths are ⅛ in. No drain holes are drilled in the groove, but twelve 5/32 in diameter holes are drilled from a relief immediately below it. When three compression rings are employed, either the top one or the top two may be chromium plated, and they are both of plain rectangular cross section, with a face width of ⅛ in. The third compression ring is stepped at its upper inner edge, thus reducing the radial stiffness of the upper portion of the ring relative to the lower portion so that it will have a slight scraper action, and to assist the bedding of the periphery during running-in. Where two compression rings only are fitted, plain rings are recommended, but the top one may be chromium plated.

Many special forms of ring and other devices have been developed to assist running-in and to prevent ring sticking. The advantage of using taper faced rings is well known. The initial running-in process occurs on the line contact round the greatest diameter of the taper, and as wear takes place, the area of contact between the face of the ring and the cylinder wall increases. An alternative to this arrangement is the use of an up-tilted ring groove. With 0·008–0·010 in up-tilt per inch radial width, a similar effect is obtained, and in addition, oil control is improved. In order to prevent ring sticking, taper sided rings and grooves are sometimes employed, the grooves and rings being wider at their outer peripheries than at their inner ones. Because of this taper, the slightest movement of the ring in its groove will free it if sticking should occur. An example of a piston

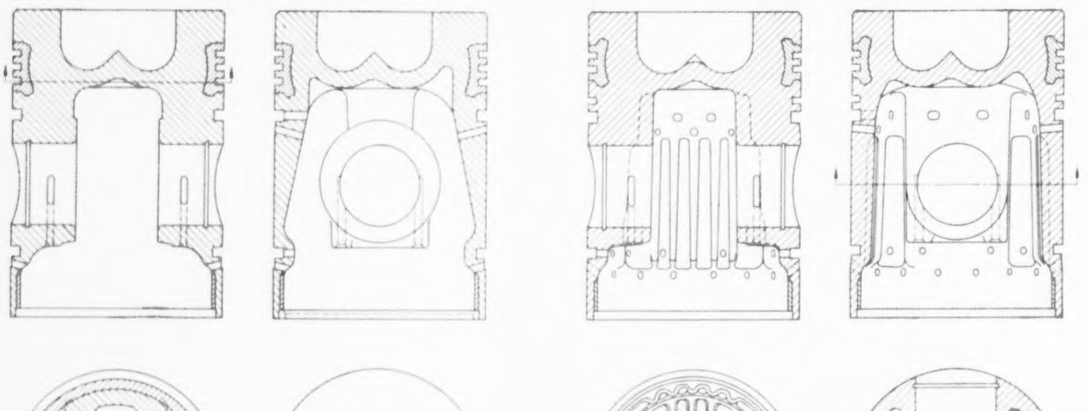


Fig. 14. Left: an early piston design in Y-alloy, with an Austenitic iron carrier for three rings, and right: a modern piston of low expansion aluminium alloy with a carrier for two rings

in which up-tilted, taper sided piston rings are used is given in Fig. 13.

It has been found that when certain diesel fuels with a high sulphur content are used, SO_3 is produced during the combustion process. Water is also formed during the process, and the SO_3 combines with it to form H_2SO_4 which, of course, is sulphuric acid. Under cold starting conditions there is a tendency for this and other corrosive vapours to condense on the walls of the cylinder. This condensation takes place because the wall temperature is lower than the dew point of the vapours which, under conditions in the cylinder, may be fairly high. Temperatures at the inside of the top of the cylinder walls should rise to above 120 deg C as soon as possible after starting. The time required for warming up is generally reduced by thermostatic control of the coolant flow. The condensed vapours have a highly corrosive action that is usually most marked in the top ring groove. Under hotter running conditions, another detrimental action takes place. This is the formation of a hard abrasive ash which again tends to collect in the top ring groove and cause wear. From the point of view of wear reduction the optimum temperature on the inside surface at the top of the cylinder walls is about 140 deg C.

The Specialloid Roebuck Armouring ring is a recent development to combat these two types of ring groove erosion. It was shown for the first time at the last Earls Court Motor Show. The manufacturers state that in one engine, after 250 hours running, the wear in the top groove without Armouring rings fitted was found to be between 0·050 and 0·055 in, but with the Armouring rings incorporated, the wear after the same period of running was only 0·003 in. The Armouring ring is simply a thin spring steel ring. Two are used, one on each side of the top compression ring or, if necessary, on each side of the other rings as well. Unlike the compression rings, the Armouring rings are sprung inwards so that they grip the base of the ring groove. Therefore, once they are pressed against the sides of the groove they remain there. Their outer edges are just below the edge of the lands.

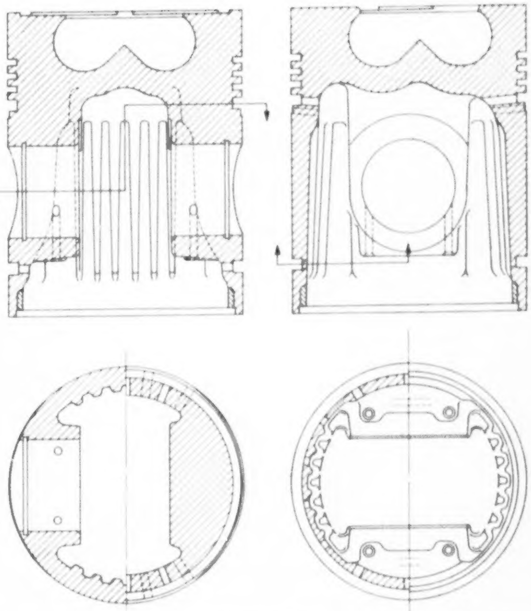


Fig. 15. A $5\frac{1}{8}$ in diameter piston with a shrouded toroidal bowl

Another method that has been adopted to combat ring groove wear problems is the use of cast-in Austenitic iron ring carriers, Fig. 14. In earlier designs the inserts carried three compression rings, but because of the fairly large temperature gradient down the insert thermal distortion occurred and the inserts tended to work loose. Later designs incorporated inserts to carry either the top ring only, or the upper two.

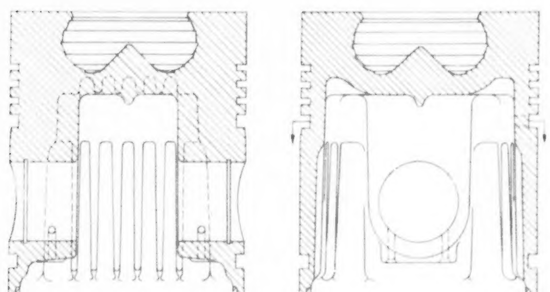


Fig. 16. A $4\frac{1}{2}$ in diameter piston with a stepped shrouded toroidal bowl

Austenitic iron inserts are generally employed in conjunction with a low expansion aluminium alloy. They have also been used with the normal high expansion alloys, successfully in many cases but not in all. The Austenitic material is used, of course, because of its high coefficient of expansion, which is not very different from that of the silicon aluminium alloys. Whether or not the use of inserts is successful depends to a large extent on the kind of service to which the engine is put. On long distance runs, temperatures are more or less constant and satisfactory piston life is obtained, but on town bus routes and local delivery services the variations in temperature experienced give rise to alternate expansion and contraction, which sometimes results in fatigue failure at the junction between the iron and the aluminium. However, a modern well designed piston with an inserted ring belt should be capable of performing any of these duties.

Requirements for various engine types

Perhaps the most severely loaded pistons are those employed in diesel engines. Although they generally run at slower speeds than petrol units, the gas loads are heavier and the temperatures higher. This means that the pistons must be of sturdier construction, and because of their greater weight, inertia loads tend to be heavier than might be expected at such slow engine speeds.

In this country, most of the diesel engines used are of the direct injection type with a combustion bowl in the crown. Any of a number of different types of bowl may be employed, for example, the open sided, shrouded, and stepped shrouded toroidal bowls, Figs. 5, 15 and 16 respectively, the open variable concave bowl, Fig. 13, and the open hemispherical type, Fig. 17.

There are advantages and disadvantages to each combustion bowl form. In general, the shrouded type gives a greater squish effect and leads to smoother running. The open toroidal and open hemispherical bowls are more prone to the onset of diesel knock, but the fuel consumption is often better. The reduced squish effect is generally offset by the use of a shrouded inlet valve which gives additional turbulence. Stepped shrouded bowls are incorporated to give even greater

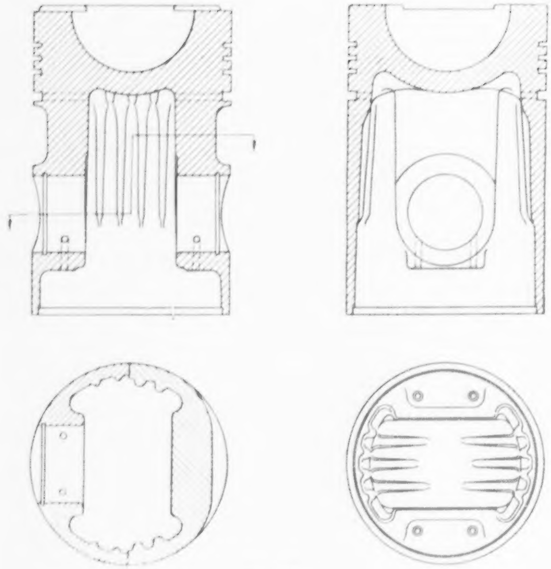


Fig. 17

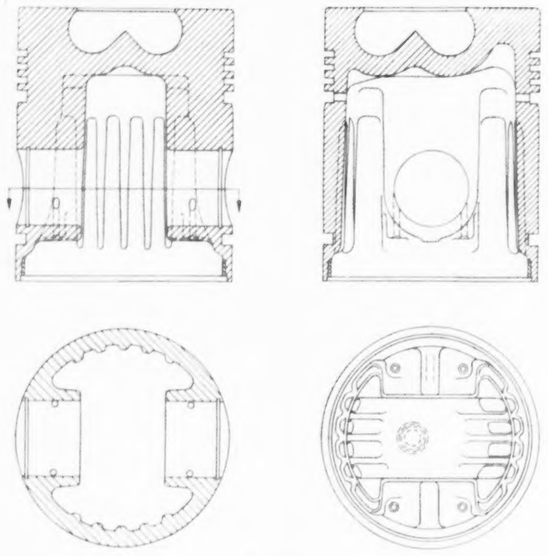


Fig. 18

Fig. 17. The Gardner $4\frac{1}{4}$ in diameter piston with an open hemispherical bowl in the crown

Fig. 18. The Sentinel $4\frac{3}{4}$ in diameter piston with the shrouded toroidal bowl offset 7·94 mm

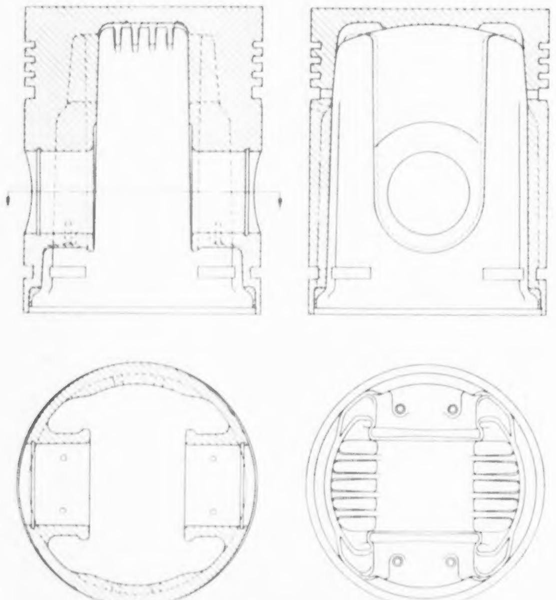


Fig. 20

Fig. 20. A 130 mm diameter heavy duty piston for the Meadows six cylinder petrol engine

Fig. 21

Fig. 21. Detonation has caused local overheating and collapse of the crown of this racing motorcycle piston

turbulence and smooth running. The open variable concave type combines some of the good features of the toroidal and the open hemispherical bowl, both with regard to its shape and the combustion characteristics obtained with it.

In most cases, combustion bowls are positioned centrally, but sometimes they are offset as in Fig. 18, in order that the inlet port may be directed in such a manner that the gas flow enters the cylinder more nearly tangentially than is otherwise possible. This increases the swirl velocity. The disadvantage of offsetting the bowl is that the resulting temperature distribution over the crown is not so even.

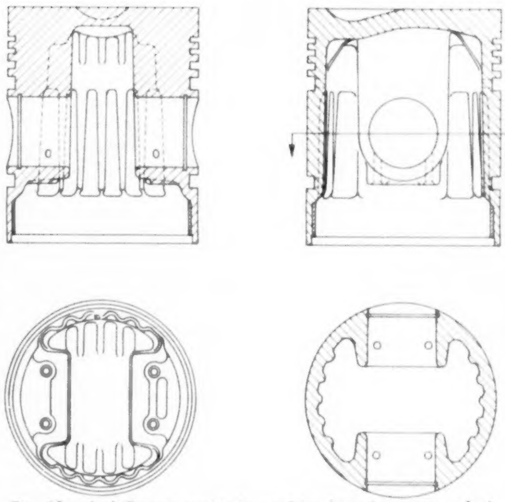


Fig. 19. A deflector cavity is machined in the crown of the 3½ in diameter piston for the Perkins range of engines

With some indirect injection engines there is a tendency towards the development of hot spots on piston crowns at the point to which the gas from the swirl chamber port is directed. The Perkins piston, Fig. 19, is rather a special case of this type of engine. The gases from the patent " Aeroflow " swirl chamber are directed towards a small deflector cavity in the crown. A two-spray nozzle is employed, one spray being directed into the swirl chamber and the other down the transfer port into the cylinder. With this arrangement, which is a combination of the direct and indirect injection systems, the local temperatures on the piston crown are appreciably reduced. It can be seen from the illustration that heat transfer webs are incorporated immediately below the deflector cavity.

Underfloor engines, which are increasing in popularity, present special problems with regard to oil control. One feature that has been introduced to assist in this respect is the deletion of the drainage holes from the oil control ring grooves on the lower side of the piston. This prevents oil from running from the inside of the piston down into the ring grooves.

In petrol engines the loads and temperatures are generally lower than in diesel units. An example of a piston for a heavy duty petrol unit, of 130 mm bore, is given in Fig. 20. This, because of its large size, closely resembles a diesel engine piston, but the skirt and crown thicknesses are not so great. Lighter petrol engine pistons, such as are used in cars usually have a split or slotted skirt. The modern tendency to square or oversquare cylinder dimensions has accentuated the oil control problem because it has brought the cylinder bores nearer to the crankshaft where they are in a position to receive more oil thrown from the crank pins. In some of these designs, deep section oil control rings are used with the result that their radial pressure is relatively high.

The principal difference between pistons for air-cooled and water-cooled engines is that the cold clearances are over 50 per cent greater in the air-

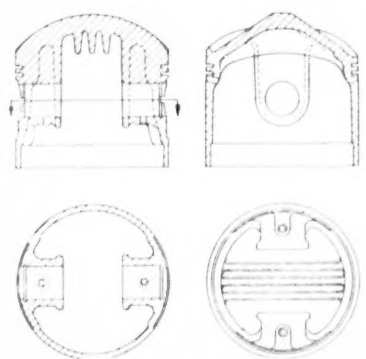


Fig. 22. A solid skirt type piston for an air-cooled racing engine

cooled units because they usually run at higher temperatures. Units of this type are usually fitted to motor cycles and are highly rated so that solid skirts are generally employed.

Racing engines for motor cars may be classified as follows: very high speed normally aspirated engines, in which piston weight must be kept as low as possible, and slower speed supercharged units in which rigidly constructed pistons must be used. In both cases strength and heat flow are of great importance. Moreover, because of the high compression ratios adopted, detonation can be a serious problem and it can burn through piston crowns in a short time, Fig. 21. Examples of pistons for high performance engines are shown in Figs. 11 (right) and 22. It will be noticed that in both these pistons, three heavy section webs are incorporated beneath the crown to support it and to assist heat flow to the skirt. Rigidity is essential particularly the part above the gudgeon pin bosses, through which most of the gas load is transmitted from the crown. Flanges, or lips, are incorporated around the inner faces of the bosses to increase the rigidity at that point; this is part of a patent specification held by Specialloid Ltd.

Until recently, Y-alloy was the material used. This was because its mechanical properties at high temperatures are better than most other aluminium alloys. However, with the introduction of improved casting techniques, notably the use of water-cooled die casting machines with a hydraulically retracted, single piece core, satisfactory results have been obtained with low expansion alloys. The thermal conductivity of these materials is not quite so good as that of Y-alloy. As a result, thicker sections must be used and this, to some extent, offsets the disadvantage arising because of the lower strength of the material. The increase in weight is not so marked as would at first appear because this alloy has a relatively low specific gravity. Forged or pressed pistons are sometimes used for racing engines, but in many cases they are not regarded as absolutely necessary.

The piston rings used in this type of engine usually have a narrow face, as little as $\frac{1}{16}$ in to reduce friction and inertia effects. In order to ensure that the gas seal is adequate, they must have a radial thickness of about D/26. They are made mainly of material to the D.T.D. 485 specification. In light weight, high performance pistons of this type, the life is restricted and they should be changed at fairly frequent intervals.

General design considerations

Contrary to earlier practice, it is now customary for engine designers to give the piston manufacturers all the relevant data concerning the engine and for the piston manufacturers to execute the preliminary design and estimate the weight. Then the proportions of the crankshaft, connecting rods and bearings can be arranged to suit the piston. This is a logical procedure since the piston has a definite and separate function to perform and must be designed specifically for it. On the other hand, the function of the crankshaft and connecting rods is to transmit loads applied by the pistons.

A list of the technical data required by piston manufacturers is given in the table. The reason for the incorporation of each individual item is in most cases apparent, but a few remarks concerning the list may help to clarify some of the less obvious points. For instance, details of the combustion system are required in order that the appropriate bowl, or other features, may be incorporated in the crown, but this information also gives an indication of the combustion temperatures and thermal stresses likely to be experienced. The type of liner employed will also have a bearing upon, among other things, the temperature of the piston. From the data concerning coolant

temperatures at maximum continuous rating, information about the temperature gradient up the cylinder bore may be derived.

Cylinder arrangement is important since it affects the tendency to distortion and the probable wear characteristics. A knowledge of the indicated horse power per cylinder is necessary because some of the development work is usually carried out on a single cylinder unit. Most of the other data included towards the end of the table is required for estimating the degree of oil control necessary. This will be affected by the method of lubricating the small end, the lubricating oil pressure, and the type of big end bearings. Larger clearances are usually used with lead bronze bearings than with other types, and if the bearings have side flanges, there is usually less oil thrown from the big ends.

So far as the overall dimensions of the piston are concerned, it is generally considered advisable to incorporate as large an area as possible above the gudgeon pin, for bearing against the cylinder wall. However, the height of the piston above the pin is limited by the height of the engine. The overall length of the skirt below the rings should be approximately equal to the diameter of the piston. In diesel engines, piston speeds are generally limited to 1600 to 1800 ft/min, while in petrol units speeds of 2400 to 3000 ft/min are common.

Acknowledgement is made to T. O. Hunt, Technical Director of Specialloid Ltd., for information which he supplied for the preparation of this article.

DATA REQUIRED FOR DIESEL PISTON DESIGN

Piston part number :
Engine type :
Intended application :
(Stationary, passenger, goods, marine, etc.)
Vertical or horizontal :
Combustion system :
Bore :
Bore limits :
Capacity :
Cylinder arrangement :
(Multi- or mono-bloc)
Type of liner :
(Chromium plated, nitrided, etc.)
Wet or dry liners :
Blown or normally aspirated :
R.p.m. at maximum continuous rating :

Rated b.h.p. :
Boost :
Compression ratio :
B.m.e.p. lb/in^2
(a) Continuous rating :
(b) Overload :
Firing pressures
(a) Continuous rating :
(b) Overload :
Indicated h.p. per cylinder
(a) Continuous rating :
(b) Overload :
Exhaust gas temperatures :
Coolant temperature at maximum continuous rating
(a) in :
(b) out :

Lubricating oil temperatures at maximum continuous rating
(a) in :
(b) out :
Fuel consumption :
(Estimated or actual)
Lubricating oil consumption :
(Estimated or actual)
Method of lubricating small end bush (splash or pressure feed) :
Big end bearings :
(Strip type without side flanges or shells with side flanges)
Lubricating oil pressure, and position at which it is taken, i.e., at pump, main gallery, or overhead rocker gear:

AIR-PRESSURE HYDRAULIC BRAKING

Clayton Dewandre Developments

ALTHOUGH hydraulic brake actuation gives a high mechanical efficiency, it does not of itself provide energy to assist the driver's effort. For applications requiring boost, the vacuum servo has been developed to form a combined unit with the hydraulic master cylinder. This gave increased brake efficiency and for petrol engined vehicles no cost was involved in providing the vacuum power. For a diesel engine vehicle, however, an exhauster is required to produce the vacuum, with a vacuum reservoir to provide power storage.

Frequently, air pressure braking equipment is specified for a vehicle with hydraulic brake transmission, and to meet the needs of such cases, Clayton Dewandre, Titanic Works, Lincoln, have developed the "Airhydro" master servo which, combined with a compressor and reservoir, replaces the vacuum equipment. The chassis builder is free to supply the vehicle with either vacuum or air pressure brakes, without having to change the brake layout.

The "Airhydro," which is similar in design and principle to the vacuum hydraulic master servos, converts air pressure into hydraulic pressure, so combining the advantages of hydraulic brake operation with an air pressure system. The unit can be made available with either a Lockheed or Girling hydraulic master cylinder. When it is used in conjunction with a tandem master cylinder, the safety factor of independent operation of front and rear wheels is retained.

At present, production of the "Airhydro" range of servos is limited to three basic designs, with 2 in, 3¾ in and 5 in air cylinder bores, produced for light, medium or heavy civilian and military vehicles. The various types and sizes of the hydraulic master cylinders are determined by the hydraulic brake system.

Because of its relatively high operating pressure, the "Airhydro" is more compact than a vacuum servo. For example, the smallest unit with a 2 in bore has an output equivalent to that of a 4½ in bore vacuum servo. With an operating pressure of 100 lb/in^2 an input effort to the servo of 445 lb through the pedal linkage produces hydraulic line pressures from 750 to 1400 lb/in^2 over the range of servos, with the exception of the 2 in bore unit in conjunction with a 1¼ in diameter master cylinder, which gives 500 lb/in^2 hydraulic pressure at an input effort of 250 lb. This point in the output performance is referred to as the knee point; it is the limit of power assistance, beyond which all additional effort to the brake pedal is transferred direct to the hydraulic master cylinder.

In the event of failure or should the vehicle be on tow, the reservoir maintains sufficient air pressure for several brake applications. When this is exhausted, the driver's effort on the brake pedal is transmitted through the servo to the hydraulic cylinder, although, of course, a greater effort will be required.

The installation of the equipment is comparatively easy and a certain degree of latitude is permissible in the positioning of the units. The "Airhydro" embodies the well known Dewandre reaction control characteristics. This ensures that the hydraulic line pressure is always proportional to the effort applied at the pedal, so that the driver has a sense of "brake feel."

A servo input rod, connected to the brake pedal, is attached to the Dewandre lever assembly, which operates through a relay lever to the distributor valve mounted on the side of the cylinder casing. When the brakes are applied, the distributor valve seals off the atmospheric connection and then admits air pressure to the power cylinder. The air pressure acting on the servo piston applies force through the lever assembly to the hydraulic cylinder, thus assisting brake application. As the servo attains an output proportional to the driver's balanced effort, the distributor lever closes owing to the balancing action of the Dewandre levers.

HIGH PRESSURE FUEL PIPES

An Investigation by C.A.V. Ltd. into Failures on C.I. Engines and their Prevention

HIGH pressure fuel pipe breakages have been experienced with many, if not all, makes of diesel engine from time to time. Investigation into causes has been carried out in the past without, however, producing any fully satisfactory solution to the problem. The frequency of failure varies greatly both from engine to engine and from operator to operator, from a pipe life of three years or even longer to frequent failure after a few weeks. Although one or two types of engine appear to be clear of the trouble so far as a few operators are concerned, there seems to be no engine that can be said to be free from pipe failures everywhere. In response to enquiries, a number of users, when first approached, said that the trouble was unknown, but subsequently found that in actual fact breakage was surprisingly common.

Failure usually consists of a break, more or less transverse, generally at the nipple, but with some terminations other types of failure, causing leakage rather than breakage, occur first. Past work has been done on the fatigue strengths of the normal single-taper pipe ends, some other forms of termination, and on pipes of some different materials. Some improvement of fatigue strength was found possible, but uncontrolled trials in service did not appear to show any considerable improvement in service life.

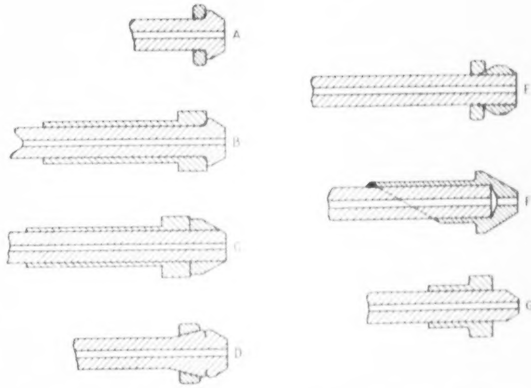


Fig. 1. Terminations for C.I. engine high pressure fuel pipes

While C.A.V. Limited has supplied fuel injection equipment for a very wide range of engines, the high pressure fuel pipes have been supplied generally by the engine builder. However, it seemed logical for the company to investigate the whole problem to try to find a means of eliminating pipe failures, and accordingly a programme of work was put in hand. A review of service practice and experience was first made, to discover if there was any fully satisfactory solution already in being. Concurrently, a laboratory investigation was made, the primary cause of failure was established, and a number of possible cures examined. As a result, a solution was devised which appears to be entirely satisfactory and generally applicable.

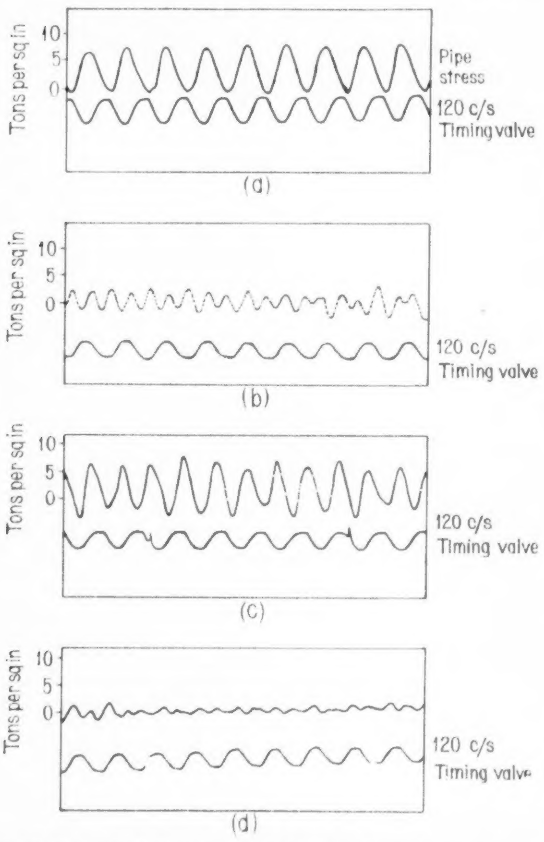


Fig. 2. Records of vibrations in a fuel pipe under various conditions

Present practice

High pressure pipes are almost invariably of mild steel and, in engines of up to 10 litres, or thereabouts, of 6 mm O.D. There are many different terminations, however, some of which are shown in Fig. 1. Of these (A) to (E) have been widely used, and there are a few others not in common use and not illustrated. The commonest termination (Fig. 1(A)) is the single taper made by upsetting or swaging with a hand tool, with a steel backing washer.

The double cone end has generally, but not invariably, been found to be an improvement over the single taper as regards breakage, but with some danger of trouble due to washer failure and consequent leakage. The loose brass olive was sometimes found to reduce breakages but with an increase of leakage trouble. Various forms of pipe clamping have been tried and many are used. They are generally found to improve matters, but seldom, if ever, to afford complete freedom from pipe failure. A number of alternative pipe materials, including Monel metal, Tungum and stainless steel, have been tried, with some improvement in the case of Monel, but without providing a complete cure.

Laboratory investigation

Most pipe failures examined appeared to be fatigue fractures due to transverse bending. Calculations of the magnitudes of some of the quantities involved are of interest. If a high pressure pipe 20 in in length is held firmly by the pump at one end and the nozzle holder at the other, a displacement of the middle of 0·1 in produces a stress at the ends of about 10 tons/in². This is the fatigue limit of the normal single cone terminations. The lateral displacement of one end of the pipe relative to the other, which would produce this stress is about 0·4 in, but vibration of the pump relative to the engine of this amount is, of course,

TABLE I

Material	End	Fatigue limit T tons/in²	Damping factor % energy loss/cycle after running at stress T′ d	T′ tons/in²	E 10^3 tons/in²	T′d/E
Mild steel	C.A.V.	9·6	0·07	9·0	13·5	4·8
	Double cone	12·4	0·07	8·0		4·2
Hard nickel	C.A.V.	9·0	0·06†		13·0	4·1
Soft nickel	C.A.V.	9·5	0·095	8·0	13·0	5·8
Halfhard Monel	C.A.V.	12·0	0·05	9·0	11·0	4·1
Soft Monel	C.A.V.	10·5	0·03	7·0	11·0	1·9
Everdur	C.A.V.	10·0*	0·018†		6·8	2·6

*Maker's value.

†These materials were not run to stabilize the damping factor since the initial values quoted here are too low to be of interest.

inconceivable. A high pressure fuel pipe, clamped at each end, is a highly resonant system, however, and if there is a vibration of one end relative to the other of the same frequency as the natural frequency of the pipe, stresses greatly in excess of those corresponding to the amplitude of the exciting vibration can be built up.

An amplification factor of 60 is quite possible, so that for a 20 in pipe, a vibration of one end relative to the other of 0·007 in would be enough to excite a vibration of amplitude 0·1 in at the middle of the pipe, producing a stress equal to the fatigue limit at the ends. Such a vibration must be reckoned reasonable. The amplification depends upon the damping which is discussed later.

It can be shown that the frequencies of pipes and probable engine or pump frequencies are roughly the same. The lowest natural frequency of a 20 in pipe in transverse vibration is 100 cycles per second. This is equal to the pump cam frequency (or engine firing frequency) of a 6 cylinder engine at 2,000 r.p.m. It seems probable, therefore, that pipe oscillations are excited by even the fundamental of the engine firing frequency, and certainly by low harmonics of it.

The stresses actually occurring in pipes were explored by strain gauge measurements on a typical 6 cylinder D.I. engine. The gauges were attached to the pump ends of the pipes, as failure occurs more frequently at this end, and in any case the stresses at the two ends are not likely to differ greatly. Results are shown in Fig. 2. (a) is a record of the vibrations resulting from plucking a pipe with the engine not running. The very slow decay of the oscillations is an indication of the very low damping of the system, showing that resonances of large amplitudes are likely.

(b) is a record taken at a speed chosen at random to indicate the general level of stress in the pipe; at certain speeds, however, much larger amplitudes are found.

(c) shows the most severe resonance observed on this particular engine. The fact that the vibration is resonant is demonstrated in (d), which shows the effect of laying a finger lightly on the middle of the pipe. This small amount of damping was enough to reduce the stress many times.

The largest stress observed on this engine was about ±6 tons/in². This is well below the fatigue limit of the single taper ends (10 tons/in²), but it is to be expected that the actual magnitudes of such resonant vibrations will vary greatly from engine to engine and condition to condition. Even this installation, therefore, cannot be said to be out of danger. Several engines have been examined, and maximum stresses ranging from 5 to 20 tons/in² have been found. This points to the conclusion that resonant vibrations leading to bending stresses reaching the fatigue limit of the pipe ends are the primary cause of failure.

The following quantities determine the stress :

1. Amplitude and frequency of the exciting vibration, that is, of pump with respect to engine.
2. Fatigue limit T of the pipe with chosen end fitting.
3. The damping factor d of the pipe at its end fitting, which is defined as the energy loss per cycle divided by the total energy of the vibration.
4. Young's modulus E of the pipe material.

Items 1, 3 and the pipe geometry determine strain, and stress is equal to strain×E. The relative merit of any given material and termination is determined by the expression Td/E.

Whatever else is done, it is of the first importance that the mounting of the pump on the engine should be as rigid as possible, since other things being equal, pipe stresses are proportional to the amplitude of vibration of pump with respect to engine. In practice, however, it would appear to be seldom, if ever, possible to achieve so rigid a system that pipe vibrations do not occur.

The possible means of reducing failures are :—

1. By increasing the fatigue limit of the pipe material and/or the termination.
2. By increasing the damping capacity of the pipe material. Measurements of both damping capacity and fatigue limit have been made.
3. By imposing restraint on the pipe, for example, by clamping.

Fatigue limit of pipe materials and terminations

Measurements of the fatigue limits of pipes of various materials with different terminations were made. A simple test rig was used, in which a short length of pipe, held by the termination under test, was subjected to reverse bending by loading as a cantilever and rotating (the Wohler test). The results are given in column 3 of Table I. The expected fatigue limit of the normal mild steel pipe itself is 12 tons/in². Thus, the double cone end is in fact slightly stronger than the pipe itself, as is shown by the fact that such ends generally break ½ in or so away from the nipple.

The brazed terminations were also fatigue tested and it was found that, when properly made, they were as strong as, or stronger than the pipe itself. Although the single cone end is slightly weaker in fatigue, the loss compared with double cone or brazed ends is small, and is shown later to be trivial compared with variations in damping factor.

The effect of the termination is somewhat greater with other materials, presumably because they are more notch sensitive. Thus, with stainless steel there is an increase from 14 tons/in² for the single cone end to 22 tons/in² for the double cone end. If such materials were to be used it would be worthwhile also using the double taper end. Apart from stainless steel, no material tested shows any considerable advantage in fatigue limit over mild steel.

TABLE II

Pipe	Unclamped Worst Resonance		Clamped Worst Resonance	
	Speed	Tons/in²	Speed	Tons/in²
1	1475	11	1200	2
2	1630	8	1800	2·5
3	1290	7	1900	3
4	1440	7	1800	2
5	1450	4	1600	1·5
6	1580	11	1500	4

Damping factor

The damping factor increases with stress and with the amount of cold working to which the test piece has been subjected; but if run at a stress near the fatigue limit the damping factor decreases with time. The relevant quantity is the stable value after running at just below the fatigue limit. Measurements under this condition have been made for most of the promising materials.

In order to avoid failure, it was generally found necessary to run at a stress appreciably below the fatigue limit, probably because of occasional increases of amplitude caused by variations of the mains supply to the test equipment. In Table I, the highest stress at which it was found possible to run is given in column 5 (T′) and the final damping factor (d) is given in column 4. The merit of any given material and termination is given by the factor T′d/E in column 7.

Some tests were also made on stainless steel. Earlier fatigue measurements gave a high strength (22 tons/in²) with double cone ends. Later samples from a different batch, however, made to measure damping factor, failed to give this high fatigue limit. Attempts to repeat the original behaviour have so far failed, but there remains a possibility that stainless steel would show a worthwhile advantage over mild steel. Apart from this possibility, the value of T′d/E given in the table shows that no other material is considerably better than mild steel. Attention was therefore given to the reduction of working stresses by means of dampers, the possibility of which had been clearly demonstrated by the effect of merely laying a finger on a pipe, described above.

Pipe dampers

Strain gauge measurements were made with various types of clamp on an engine which gave pipe stresses, without clamps, of up to 11 tons/in². It will be appreciated that in a system such that the high stresses are due to resonance, the stresses found may be capricious from one occasion to another. It is therefore necessary that for any device to be regarded as offering a cure for the breakage trouble, it must give stresses very substantially less than the fatigue limit; the target set in these experiments was one half the fatigue limit, i.e., with single cone ends on mild steel pipes, a stress not greater than 5 tons/in².

The arrangement most likely to be effective is one in which each pipe is connected to the engine structure by some material capable of absorbing vibrational energy. It was found that attaching the mid-point of each pipe to the engine by means of a clamp lined with sponge rubber was very effective, provided the clamp was not fully tightened. With the sponge-rubber lined clamp fully tightened, however or with rigid (steel) clamps to the engine, the particular resonance giving the highest stress, with pipe unclamped, disappeared, but resonances appeared at other engine speeds which were generally almost as serious as the original one.

A similar result was obtained by clamping the pipes together in pairs with either steel, wood-lined steel, or even sponge-rubber lined clamps not fully tightened. It was concluded that of these palliatives the most effective was clamping to the engine with resilient, energy-absorbing material interposed.

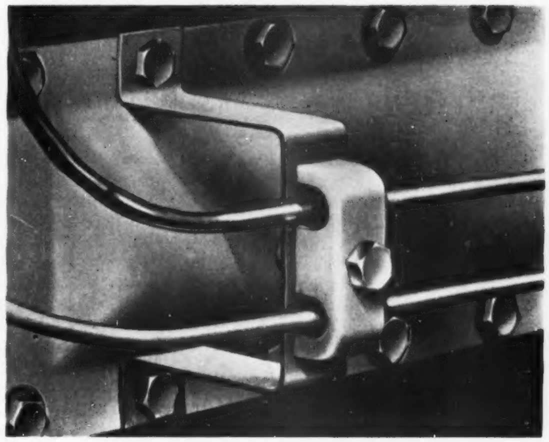


Fig. 3. A C.A.V. double pipe damper

The C.A.V. synthetic rubber damper (Fig. 3) was then designed. It consists of a rubber bush which fits the pipe, the outside surfaces being held in a metal housing which fits but does not deflect the rubber. The effect of these dampers in reducing stresses is shown in Table II. Subsequently, tests were made on two other engines with similar results, an occasional stress of 4 tons/in² being observed on one pipe. Fig. 3 shows a twin pipe damper.

Endurance tests were made in which the rubber was stressed much more severely than it would be in practice, by vibrating a pipe in it. Runs were made with the damper dry at room temperature, and at 80 deg C with gas-oil dripped on it. Only inconsiderable wear was found in 100 hours, and in view of the much more severe duty which rubber components often perform, it is expected that these dampers will last many thousands of hours; it is proposed, however, that the rubber should be replaced at engine overhauls.

Recommendations on pipe installations

1. The pump mounting and the part of the engine structure to which it is attached should be as rigid as possible.
2. Pipes should be clamped at about the mid-point to the engine structure. Where, to keep pipes of equal length, it is necessary to put a coil into a pipe, the coil should be neglected when estimating the mid-point.
3. The layout should be such that the pipe is as far as possible parallel with the axis of the damper. If a major part of either half of the pipes makes an angle greater than 45 deg with the axis of the damper, the damper may be ineffective.
4. The pipe should be smooth at the point when it goes through the damper.
5. In forming the pipe ends care should be taken to avoid notching the material. Flashes should be removed only if essential to permit assembly; if they must be removed great care should be taken to avoid gashing with the file.
6. Care should be taken when forming nipples with a hand operated nipple forming tool, that undue pressure is not applied as this distorts the nipple and reduces the bore of the pipe.
7. After formation of nipple and setting to required angle, pipes should be cleared either by blowing through with compressed air or with fuel or Shell Fusus 'A' oil under pressure.
8. If pipes are to be stored, ends should be blanked off to exclude air, or pipes submerged in a tank containing fuel or Shell Fusus 'A' oil.

The C.A.V. damper of the type shown in Fig. 3, is available in single double or triple pipe units.

Radio Designer's Handbook

THIS is a comprehensive reference handbook for all who are interested in the design and application of radio receivers and audio amplifiers. The work deals in detail with basic principles and practical design. This fourth edition is more than four times as large as the previous edition. It is the work of 10 authors and 23 collaborating engineers under the editorship of F. Langford Smith, B.Sc.

The enormous amount of data it contains has been made readily accessible by means of a fully detailed list of contents and a very complete index. In itself the book is a self-contained source of information, but exhaustive bibliographies are also provided. The main subjects are:—valves and valve testing; general theory and components; radio frequencies; power supplies; design of complete A.M. and F.M. receivers; and reference information. This book is published by Iliffe and Sons Ltd., Dorset House, Stamford Street, London, S.E.1, at 42s. (postage 1s. 6d.).

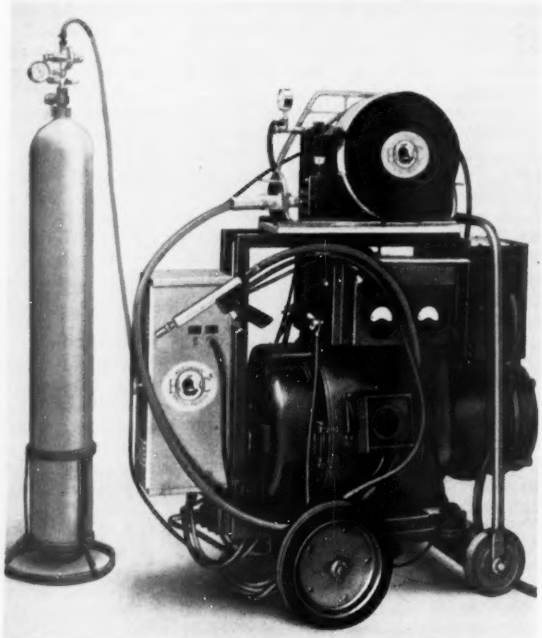
ARC WELDING

The Argonaut Manual Process

ARC welding in a protective shield of the inert gas, argon, by the Argonarc process is now a well-established practice. The originators of this process, The British Oxygen Company Limited, have now developed the Argonaut process, which is in effect a manual Argonarc process using the inert gas shielded arc but with a consumable electrode and automatic regulation. This process uses a shielded arc with direct current (electrode positive) of relatively high amperage (50,000 amp/in^2 minimum) on a continuously fed bare wire electrode of small diameter. The high current gives a very fast rate of deposition. In addition, the manner of metal projection across the arc permits welding in all positions. These two attributes give the process a large field of application on all types of fabrication where relatively heavy section aluminium is welded *in situ*.

Flux is not needed, and the welds are of good quality, free from slag and inclusions. They are also, of course, free from post-weld corrosion problems. With multi-pass techniques there is no practical limit to the thicknesses that can be welded, and comparatively few passes are needed because of the large amount of filler that can be deposited in unit time. By using smaller diameter electrode wire and lower current, thin metal down to $\frac{3}{16}$ in may also be welded. The efficiency of the process on thin work is comparable to that on heavy sections.

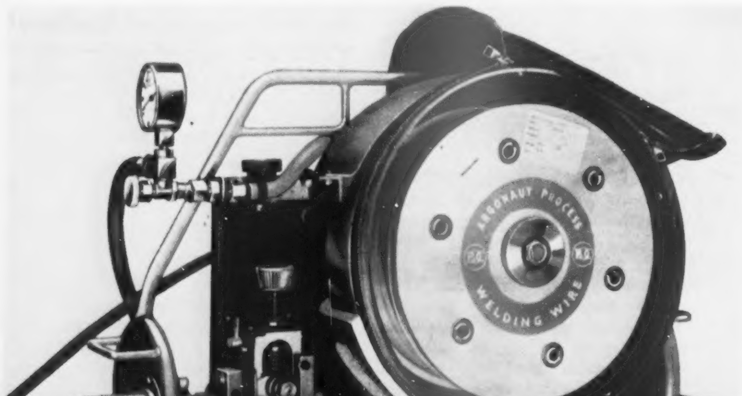
The arc is self-adjusting and the equipment includes a reel and feed motor to supply the filler wire to the torch or gun. Feed to the gun is via a special cable which also conducts the argon gas to provide the air-excluding shield round the arc and over the weld pool. Automatic adjustment of the arc is governed by the burn-off rate of the wire, which is fed at a constant speed through the gun. If the operator tends to draw the gun away from the work, so lengthening the arc, the burn-off rate is reduced and the wire feeds to the work to shorten the arc. On the other hand, if the gun is brought too close to the work, the reverse happens. Thus the operator is given a reasonable latitude without there being any adverse effect on the weld.



Argonaut unit complete with associated equipment

To start the welding, the electrode or filler wire is brought forward so that it projects about $\frac{1}{2}$ in from the nozzle of the gun. When the trigger is pressed, the welding circuit is switched on and argon starts to flow. The tip of the wire is then brought down to the face of the joint and the arc is struck. At this stage a relay starts the wire feed motor and the wire is brought forward and the burn-off action continues. The fusion rate is very much higher than with conventional arc welding and consequently a higher weld speed is obtained. The use of the process is not confined to light alloys; it may also be employed on stainless steel and copper base alloys. Certain types of welds between dissimilar metals, such as copper to steel, can also be effected by employing an aluminium bronze wire as the filler material.

From the operator's point of view, the Argonaut process is very easy to use in the downhand position for both fillet and butt welding. It is, however, for vertical and overhead welding of fillets and butts that the process scores most heavily, since operation in the more difficult positions is much easier than the comparative methods for normal manual arc welding. The most notable improvement is for welding in the overhead position, because the effects of gravity on the weld metal crossing the arc are scarcely noticeable. This allows an operator of only average skill to make welds in positions that would otherwise call for a highly skilled operator. The principal difference in technique is the necessity to move along the joint far more quickly than is customary.



The unit with covers removed to show the reel and the drive rollers for feeding the wire to the gun

TROLLEY BUS DEVELOPMENTS

A Review of Present Practice

IN the past two years there has been little change in the design of trolley bus chassis. The units of the electrical equipment, produced in similar forms by five independent organizations, show less change, except in detail, than do the mechanical parts. As a reduction of about 10 to 1 is required, underslung worm drive is generally used. The nominal power of the average motor on a one hour rating is about 100 b.h.p., and on a well-ventilated machine about 90 per cent of this is continuously available.

The design of the motor is also such that a very high overload over a wide speed range can be carried for short periods without harm. As a result the control system must be such as to limit automatically the acceleration and braking to an agreed figure, generally in the order of three miles per hour per second. This is as high as passenger comfort and safety and mechanical considerations will permit. The largest double-deck, 70 passenger trolley-bus can accelerate to 15 m.p.h. in five seconds and to 25 m.p.h. in 15 seconds.

Location of the units comprising the electrical equipment is determined chiefly by the character of the body and the purpose for which it is intended. With single deck bodies, the contactor group may be mounted at the rear and be accessible from the outside; alternatively it may be on the chassis side frame with the main shunt field resistance mounted between the frame members near to the contactors so as to limit the length of cable run. If the master controller is mounted under the driver's seat, a passenger entrance can be arranged forward of the front wheel. A second doorway can be provided in the centre of the vehicle where normal rear equipment is used, or the second doorway can be at the rear overhang when the contactors are side mounted.

For overseas application, which are subject to few restrictions on overall length, the traction motor can be mounted behind the rear axle to give a better distribution of weight. This is particularly desirable when a large seating and standing capacity is required in a single deck vehicle. The B.U.T. chassis type E.T.B.1 and the Sunbeam M.F.2R. chassis shown at the 1952 Commercial Motor Show are typical examples. The B.U.T. chassis has English Electric equipment throughout, while B.T.H. electrical equipment is fitted to the Sunbeam chassis.

Both two- and three-axle trolley buses usually have the driver's cab extending to the full width of the vehicle. This allows the contactor group to be mounted abreast of the driver and adjacent to the master controller with which it is usually combined. The main resistance is then generally mounted between the chassis side members and immediately behind the cab bulkhead.

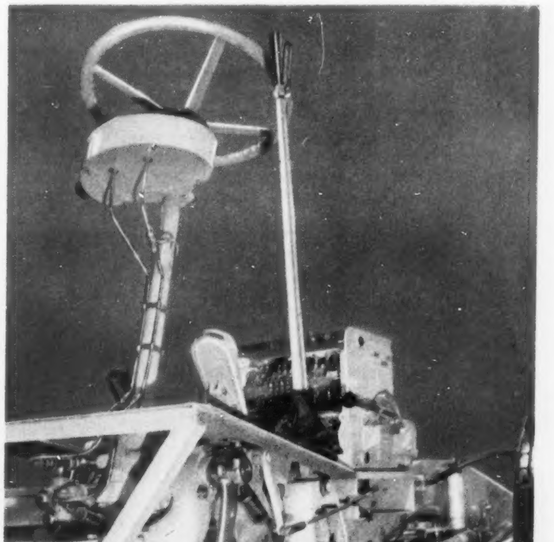
The Sunbeam M.F.2R. two-axle chassis is substantially the same as that described in the *Automobile Engineer* for January 1951. It has an overall length of 32 ft 5 in with a wheel base of 17 ft 6 in and is designed for overseas use with single deck bodies of the Transit type, having seating capacity up to 44 persons. Alternative overall dimensions and wheel bases are possible with this design to accommodate different length bodies.

The channel section frame is upswept over front and rear axles and strongly braced by pressed and tubular cross members. The front axle is a drop forged alloy steel beam, with taper roller thrust bearings on the king pins to reduce steering effort. An underslung worm drive rear axle with fully floating axle shafts is used and a single short transmission shaft with needle roller bearing universal joints takes the drive from the motor to the worm shaft. The worm gear ratio is 10·33 to 1 and all tyres are low pressure 10·00-22 type, singles being used on front and twin tyres on rear wheels. An air pressure braking system is used to operate internal expanding brakes on all wheels. It is brought into operation towards the end of the brake pedal motion since there are two stages of stabilized rheostatic braking which come into operation before the mechanical braking. A small air compressor driven by a 1·8 h.p. electric motor and having a capacity of 10 cu ft/min is adequate for both the braking requirements and pneumatic operation of doors. The unit is carried on the outer face of a side frame member. A hand brake of the double trailing pawl type with rack and pawl mechanism, enclosed to exclude dirt, operates on the rear wheels only.

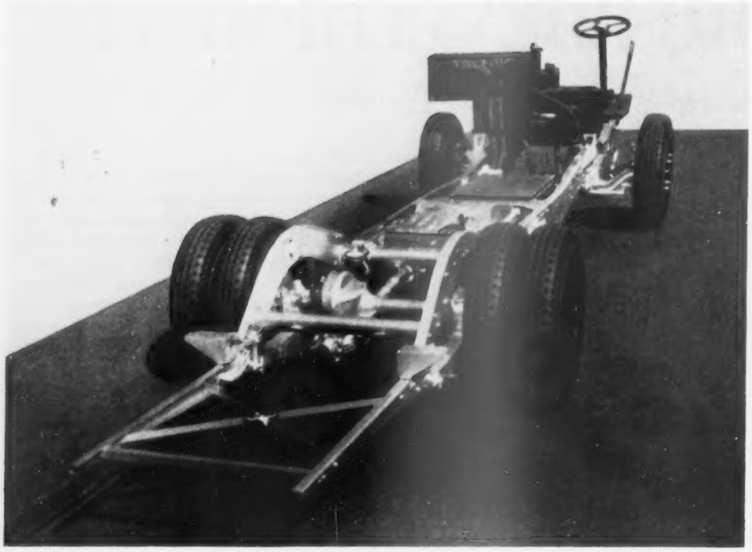
The steering and controls are forward of the front axle and the B.T.H. master controller type C.679 is mounted under the driver's seat. The steering is so designed that it is possible to use a single side steering connection from the drop arm to the axle and still obtain a small turning circle.

As before, the motor is rear mounted to improve weight distribution. It is a B.T.H. 210 flood proof model with a 30 volt overhung B.T.H. generator for interior lighting and battery charging. The generator armature is mounted on an extension of the motor shaft and the body bolted to the commutator end cover so that the machine is integral with the motor, sharing a common ventilating system and with the same flood proof feature.

The contactor panel, B. T. H. type R.M.C., is mounted in a waterproof steel cabinet at the side of the chassis frame and the resisters are carried in a frame between the side members and suitably shielded. Automatic acceleration is included and the relay panel for the control is fixed to the chassis side frame beside the main contactor panel and enclosed in a waterproof casing. All power wiring is enclosed in



Steering layout on Sunbeam M.F.2R chassis with B.T.H. electrical equipment



Sunbeam F4/A chassis

head. Automatic acceleration and battery manœuvring are included, together with all the mechanical and electrical interlocking gear required to operate the vehicle. Power wiring is enclosed in rubber or flexible metal hose, and all resisters are arranged in one unit between the side members behind the bulkhead. The unit is protected from snow and splash without interfering with the free circulation of air for cooling.

Another Sunbeam chassis, the F4, is of the two axle type and is similar in most respects to the F 4/A type. Air pressure braking is used in this model instead of hydraulic and the electrical equipment is B.T.H. throughout. The cab control unit is the B.T.H. T.B.C. type, attached to the bulkhead in the cab on the left of the driver. It contains all the control equipment with the exception of the main resisters, which are shielded from weather conditions and carried on insulated supports in two sections between the side members of the chassis frame. The traction motor is the B.T.H. 209 floodproof type of 95 h.p. mounted in front of the rear axle which has an underslung worm drive with fully floating axles, as in the F 4/A model. The gear ratio is 9·66 : 1, the wheels and tyre sizes being the same as on the F 4/A.

Normally, the control system has ten accelerating and two braking steps or notches, but for difficult country the accelerating steps can be increased to thirteen and the braking to three, either for stabilized rheostatic or contra field braking. The latter system is used where there is any risk of snow getting on to the resister insulators with the consequent risk of shock to passengers. On accelerating, the first notch is sequence interlocked with the second to provide a fog running point with reduced motor current since the shunt field is energized on the second notch. In the 13 step design the starting resistance is progressively cut out up to

flexible metallic hose giving protection against chafing and road splash.

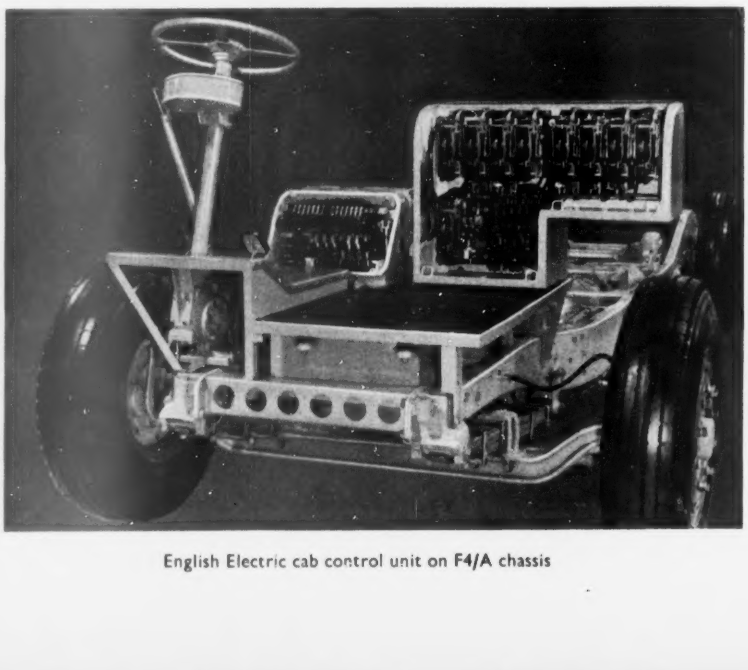
The Sunbeam F 4/A chassis is an example of the latest design by that Company of two-axle chassis. Though this chassis is intended for double-deck bodies, the overall length and wheel base dimensions can be varied to suit single-deck bodies when required. This particular chassis has an overall length of 26 ft 8 in and a wheel base of 16 ft 4 in. Alternative wheel bases of 17 ft 6 in and 18 ft 6 in are available with correspondingly increased overall dimensions. Hydraulic brakes are fitted and the electrical equipment is by the English Electric Company throughout. Special attention has been given to simplifying maintenance work and all units are readily accessible from a pit, so obviating the need for hinged inspection covers in the body sides.

A channel section alloy steel frame is employed. It is reinforced by inserts at the front and rear upswept portions and braced by pressed and tubular cross members. The front axle beam is a high tensile steel drop forging with taper roller bearings on the king pins as on the M.F.2R. chassis. An underslung worm back axle and transmission is used but the motor in this chassis is in front of the rear axle. The gear ratio is 9·66 to 1, and twin $10{\cdot}00 \times 20$ low pressure tyres are fitted on rear with $11{\cdot}00 \times 20$ single tyres on front wheels. All hubs have taper roller bearings and the brake drums are in alloy cast iron with anti-squeal bands.

Lockheed hydraulic servo brakes on this chassis use a combined power valve and cut-out, operating individual brake cylinders for each wheel. The pump is driven by twin belts from the transmission and a simple method of belt adjustment is provided. By the addition of an extra accumulator this system can be arranged to provide hydraulic power for operating doors.

Automatic brake adjusters are fitted to each wheel and the pedal pressure required to produce a given braking effort is arranged to be the same as that required by the compressed air system. The two stages of rheostatic braking are controlled by the brake pedal, which towards the end of its travel brings in the mechanical brake in addition. The hand brake operates on the rear wheels only and is of the double trailing rack and pawl type as on the M.F.2R. chassis. Marles cam and double roller steering is used and a group system of lubrication is fitted.

The English Electric equipment installed on the chassis is almost identical with that on the complete trolley bus. The cab control unit includes the master controller, located under the driver's seat, and the complete contactor group unit mounted on the driver's left and fixed to the bulk-



English Electric cab control unit on F4/A chassis

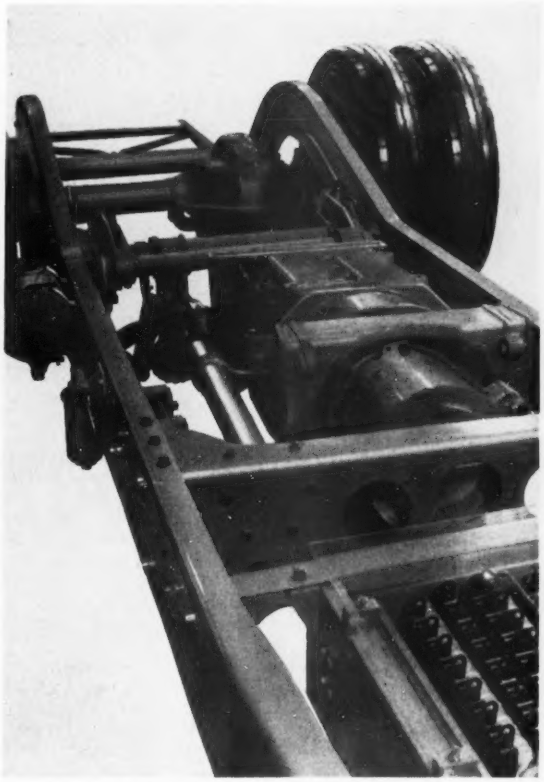
the tenth notch which gives an economic running point with full field. The remaining three notches weaken the field by first cutting out the shunt winding and then diverting part of the series winding in two stages. Thirteen stages, operated in this manner, are used where gradients demand reduced increments of accelerating torque for smoothness of operation. The automatic accelerator unit has four electrically-operated relays which allow acceleration to proceed at any rate up to a controlled maximum, provided that the motor current does not exceed the permissible value. The control is on a time basis and is also limited by motor current.

A Sunbeam M.F.2.B. trolley bus chassis with double-deck body is of special interest since it is an overseas type of chassis modified by reducing the wheelbase to 15 ft 6 in and the overall length to 27 ft to bring it within the Ministry of Transport Regulations. The vehicle is to the order of the Kingston-on-Hull Corporation.

The double-deck body is of composite construction and built by Charles Roe Ltd., of Leeds. It has seating capacity for 54 passengers, 20 in the lower compartment and 34 in the upper saloon. The body design is completely new, with separate entrance and exit doors. The entrance door is forward of the front axle and there is a central exit and two staircases to the upper saloon. This arrangement has been adopted to speed up the loading and unloading and ease the flow of passengers through the vehicle. The doors are air operated and their mechanism is interlocked with the control circuit of the trolley bus for safety.

Metropolitan-Vickers traction equipment is employed. The master controller is located under the driver's seat and the contactor panel unit, though mounted on the chassis, is inside the bodywork under the rear staircase and accessible through hinged panels in the side of the body. A 95 h.p. compound flood-proof motor carries an overhung C.A.V. generator for interior and traffic lights. B.T.H. resisters type R.P. are chassis mounted and protected against weather conditions.

The British United Traction model 9641.T. is a three-axle trolley bus chassis with a wheelbase of 18 ft 5 in to centre of bogie and an overall length of 29 ft 7 in. It is one of a fleet of double-deck trolley buses electrically equipped by the General Electric Company for Belfast Corporation.



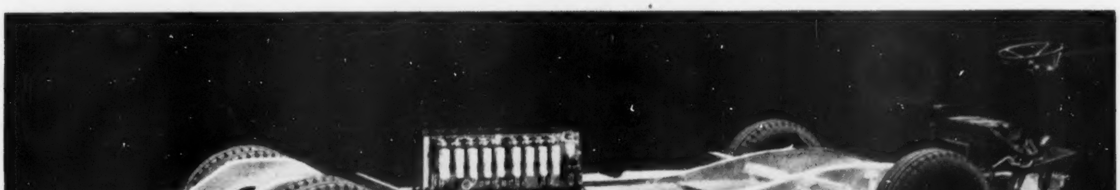
Rear end of Sunbeam F4/A chassis

The channel section frame of alloy steel has a maximum depth of 11¼ in and the tubular steel cross members are positioned to carry the electrical equipment, including the driving motor, as well as to impart the necessary rigidity to the frame. Strengthening flitches of alloy steel further stiffen the frame where it rises over the front and rear axles.

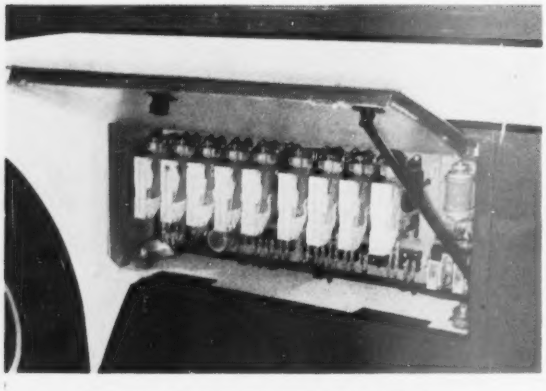
The front axle beam is an alloy steel stamping of I-section with integrally formed spring pads, and the steering swivels are of high tension alloy steel with phosphor bronze bushes at the top and bottom. Vertical load is taken by taper roller thrust bearings, encased in pressed steel housings to exclude mud and water, and carried on self-aligning spherical seats. The hollow swivel pins are of high tensile alloy steel chromium plated to minimize wear. They fit into tapered bores in the ends of the axle beam, and their retaining nuts also hold the front air brake cylinders.

Worm and nut steering is employed with the rocker shaft carrying a bronze nut that swivels on two trunnions and is operated by a thread cut on the lower end of the steering shaft. The upper end of the shaft is carried in a duplex double purpose ball bearing. Worm driven rear axles with fully floating axle shafts are used, a third differential being embodied in the drive between the two axles forming the rear bogie. The gear ratio is 10¼ : 1 with an alternative of 9¼ : 1; the tyre size is $11{\cdot}00 \times 20$.

Single inverted semi-elliptic springs at each side are pinned at their ends to the two rear axles forming the bogie and are pivoted to the frame through large diameter plain bearing trunnions and cast steel brackets bolted to the side members. Horizontal radius links



Sunbeam M.F.2B chassis with Metropolitan-Vickers electrical equipment



Contactor panel on the Sunbeam M.F.2B chassis

transmit the torque reaction directly from torque arms on the axle casing to a bracket bolted to the side frame, thus ensuring equal loading of the two axles under all conditions of acceleration and braking.

The motor is attached to the chassis frame by resilient mountings. Drive to the first rear axle is by a short propeller shaft with Hardy Spicer wide angle couplings at the front and rear. Inter-axle drive is by a similar Hardy Spicer shaft.

Compressed air braking is used. It operates internal expanding brakes on all wheels and is brought into action towards the end of the brake pedal motion after two stages of stabilized rheostatic braking which come into action before the mechanical braking. The compressor, which has a displacement of 4·5 cu ft/min is driven by a ¼ h.p. motor and maintains a pressure of 70 to 90 lb/in^2 in its 0·75 cu ft reservoir. A pull-on type hand brake with a five pitch ratchet and pawl operates the eight shoes in the rear axles independently of the air brake system.

G.E.C. electrical equipment is used throughout. The master controller and the complete contactor unit are mounted together inside the cab, the former being located under the driver's seat with the contactor unit fixed to the bulkhead on the driver's left. The master controller is of the silver butt contact type and contains the sections for operating speed control, automatic acceleration, rheostatic braking and also includes the reversing drum. Mechanical interlocks prevent power being applied when braking is in operation or unless the reverse drum is in its correct position; the reverse drum cannot be moved unless both control sections are in the "off" position.

Automatic acceleration is obtained through hydraulic dash pot or retarder and a spring in the coupling rod between the power pedal and the master controller. The retarder has two valves, one of which—the main valve—is controlled by a magnetic relay operated by the main motor current. As long as this current remains below a predetermined value the valve is held open to allow the master controller to follow the pedal. Any rise in motor current closes the valve immediately and arrests the controller's progression. To obviate stalling on hills or in similar circumstances, the second valve in the retarder, which is independent of the current control of the main valve, allows the master controller to follow the pedal at a slow rate. The maximum rate of acceleration is thus limited to a predetermined value, in this case about 2 miles per hour per second. Any rate lower than this is under the driver's control. The contactors are of the clapper type arranged for mounting on the front of the panel with front connections. Interlocks actuated by the contactors through bakelite links are fitted as required.

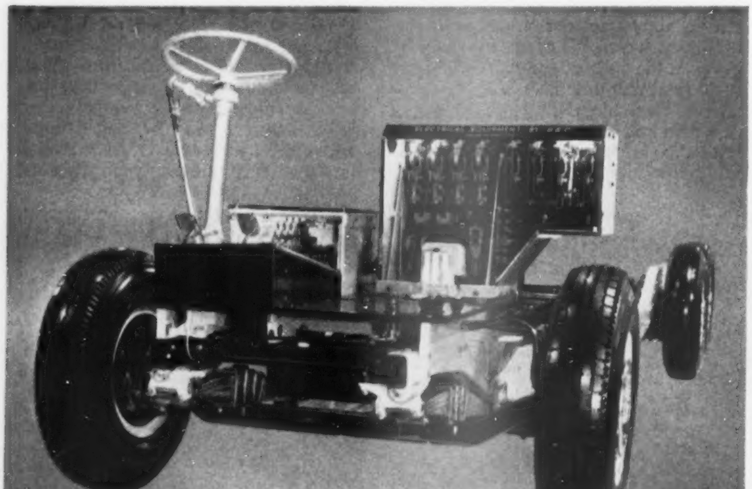
Protection to the electrical equipment is provided by two circuit breakers, one in each pole, mounted together in one case in any convenient position within reach of the driver. The motor is also protected by an overload relay which prevents the breakers from opening if a momentary overload occurs. It also allows the driver to re-apply power after first bringing the controller back to the off position. To prevent interference with radio reception, suitable condensers are connected across the H.T. circuits with choke coils in the positive and negative control feeds.

A flood-proof compound wound motor rated at 125 h.p. is used. Attached to the commutator end is an overhung C.A.V. voltage controlled generator of 1440 watts output at 24 volts nominal. The cutting in speed is about 3½ miles per hour. It supplies the interior and traffic lights in conjunction with a normal size battery. Battery manœuvring is not required.

The control of the driving motor is such that it operates as a compound wound motor for the first six control steps, parts of the starting resistance in series with the armature being progressively cut out at each step. On step 7 it is running as a compound wound motor at line voltage with all resistance cut out. On step 8 the shunt field is cut out, the motor then running as a series motor, and on the last two steps a part of the series field is progressively diverted. There are two stages of electric braking in which a part of the series field is used to oppose the shunt field and so limit the maximum torque to a safe value. The last part of the pedal motion brings in mechanical braking.

The British United Traction chassis type E.T.B.1 is a two axle model designed for single deck trolley bus bodies and is one of a fleet with electrical equipment by the English Electric Company for Glasgow Corporation. It differs slightly from the standard B.U.T. specification in that the overall length has been reduced to 29 ft 2½ in to give a complete vehicle length of 30 ft 0 in. Those dimensions have been obtained by reducing the wheel base from 17 ft 6 in to 15 ft 7½ in and shortening the overhang at the rear. This overhang has been raised so that a flat floor can be provided running throughout the length of the vehicle interior.

The chassis frame side members are of channel section alloy steel upswept over the front and rear axles and tied



General Electric cab control unit on B.U.T. 3-axle chassis

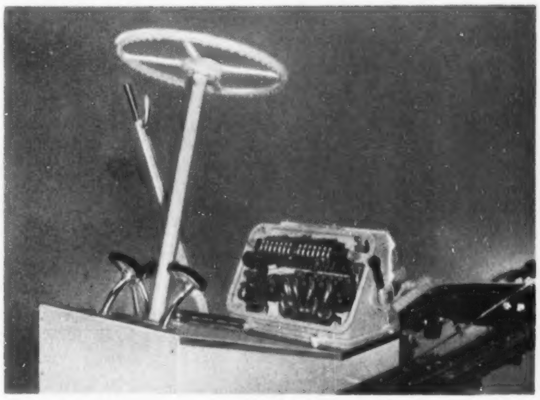
together by one channel section and eight tubular cross members. The traction motor is carried between the left-hand side member and an additional short tubular longitudinal member between the two cross members ahead of the rear axle. Resilient mountings are used and the drive to the axle is by a short propeller shaft with needle bearing universal joints. The rear axle is of the fully floating type with underslung worm drive of 9¼ to 1 ratio. 11·00 × 20 low pressure tyres are fitted all round with twins at the rear. Marles cam and double roller steering is fitted, the motion being transmitted by an intermediate drag link to a relay shaft and thence by a second drag link to the axle swivel so that a large angle of lock is obtained. The foot brake operates on the Lockheed hydraulic accumulator servo system and a special valve is used to give a pedal pressure in operation similar to that of an air brake system.

English Electric equipment installed on this chassis is of the same basic design as that on the Sunbeam chassis type F 4/A. In the case of the B.U.T. chassis, however, as it is intended for a single-deck body, the contactor group and field resistances are mounted on the side frame of the chassis and the master controller under the driver's seat, so leaving room for a passenger entrance forward of the front wheel. In the two Sunbeam installations fitted with this equipment the master controller and contactor panel are incorporated in the cab control unit bolted to the front bulkhead.

All electrical contacts are of deoxidized silver and of the butt type, designed to have a mechanical wiping action to prevent corrosion. Mechanical interlocks are fitted so that the reverser cannot be moved with either the power or brake pedal depressed or the power pedal moved with the reverser at the "off" position. Battery manœuvring is provided and the change-over from trolley to battery operation is made by contacts on the reverser. There is, in addition, a series-parallel battery switch and contactor connecting the series batteries with the motor and operated by a foot switch. The traction motor is the 120 h.p. type 410 flood-proof model with C.A.V. overhung generator for low tension lighting. It is carried on bonded rubber mountings.

The standard English Electric method of control as used on the three installations referred to is the S.D. (Series-Dynamic) system in which the series field only is used for accelerating, giving the steep gradient characteristic required to minimize current peaks on each notch. Use of the shunt field for braking only and the series field for power only permits a small shunt winding to be used, and as the motor field is at a minimum when the rheostatic braking circuit is established and there is only residual voltage on the motor, the braking current is built up gradually as the shunt field builds up.

Eleven steps are provided for acceleration. The series field only being used, it carries the full armature current. At each of the first eight steps the resistance in series with the motor is cut out progressively so that on the eighth step the motor is connected directly across the line, all the resistance having been cut out. The three remaining steps are given by progressively weakening the series field by connecting diverting resistances in parallel.



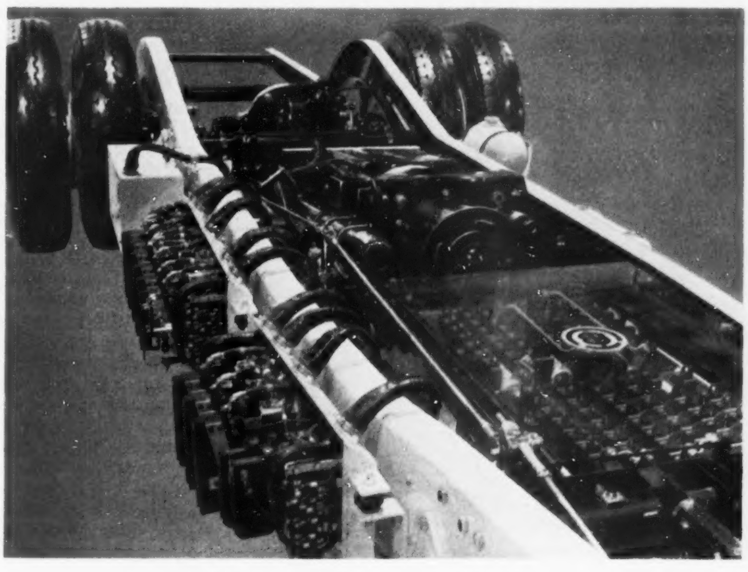
English Electric master controller on B.U.T. 2-axle chassis

For braking, the shunt field is connected across the line in series with a resistance which is cut in two stages, the armature being connected to a section of the main starting resistance. To prevent the maximum braking torque exceeding the safe working load on the rear axle, a differential effect is obtained by connecting the shunt field to the line through a section of the main resistance carrying the armature current. The potential across this, due to the current, is used to oppose and limit the shunt current, which in turn limits the rise in braking current and stabilizes the braking effort at each of the two steps. Further depression of the brake pedal holds the electric brake and brings in the hydraulic brake in addition.

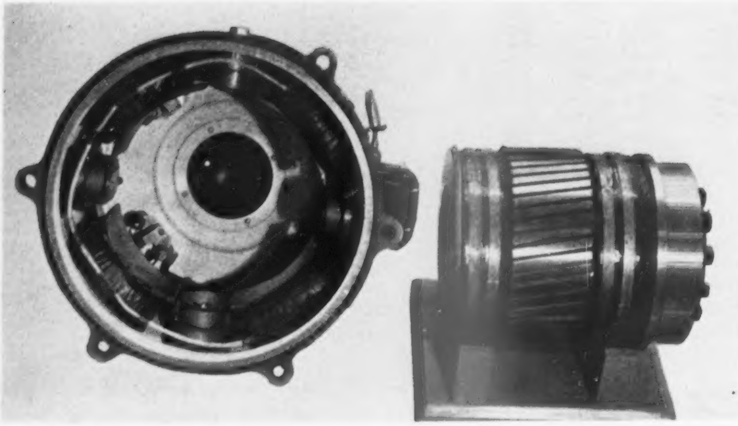
The automatic accelerator consists of a unit containing hydraulic valves which are under the control of a spring-loaded rod from the power pedal. A current limit relay set at a predetermined value of current, controls an electro-magnetically operated stop valve. If the current rises above this value the relay operates to prevent further progression of the controller until the current value falls to the level required for the current limit relay to open and release the stop valve. The timing of the operation of the controller can be adjusted to meet service requirements. The current limit relay is mounted on the main contactor panel and the hydraulic unit is fitted on the back of the master controller.

Radio interference is prevented by fitting a small choke in each main lead and there is also a tuned condenser filter connected to the trolley leads, which acts as a low-impedance shunt between them and the chassis at radio frequences. Since the condenser filter is connected to the main trolley leads, it also suppresses interference from the main motor, compressor set and motor generator set when fitted.

The C.A.V. overhung generator type D 13 T.B. is a shunt wound four-pole machine fitted with commutation poles



English Electric equipment on B.U.T. 2-axle chassis



Field and armature for C.A.V. overhung generator

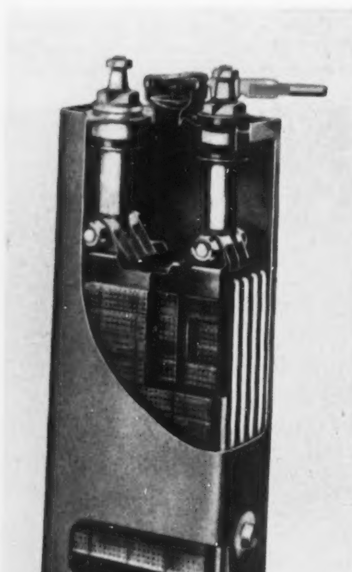
and having a nominal output of 1500 watts. It can be supplied with a 24 or 30 volt winding as required. The armature is built on a steel sleeve for attaching to an extension of the traction motor shaft, the yoke, pole and brush gear assembly being attached to the motor end shield by a spigoted flange. This machine is designed to have a low cutting-in speed and a wide speed range. Commutation poles are fitted to ensure sparkless commutation over all conditions of load and speed.

The 24-volt machine cuts in at 270 r.p.m., corresponding to a road speed of 3·1 m.p.h. with a 10·33 axle ratio. Full lamp load of 35 amps is carried at 420 r.p.m., corresponding to 4·9 m.p.h., and the maximum output of 60 amps is reached at 520 r.p.m., or 6 m.p.h. The corresponding road speeds for the 30 volt design are 3·6, 4·3 and 5·7 m.p.h. respectively. The machine operates with the C.A.V. compensated voltage regulator and cut-out.

Alkaline batteries are now extensively used on account of their robustness and long life. They differ in many ways from the lead-acid type. For example, nineteen cells are required for a 24-volt and 24 cells for a 30-volt system owing to the lower voltage per cell as compared with the lead acid type. In spite, however, of the greater number of cells required there is a considerable saving both in space and weight in favour of the alkaline type for the same duty.

Since the introduction of the nickel cadmium to supersede the nickel iron type, the design has undergone internal developments without sacrificing the original qualities such as the ability to withstand hard mechanical treatment, including shocks and vibrations, and neglect. It is almost impossible to damage the plates through overcharging and the battery can be left in a discharged state practically indefinitely, since there is no internal action corresponding to sulphating in the case of the lead-acid type. As the electrolyte is alkaline the terminals do not become corroded and there is no local action between it and the constituents of the cell so that on open circuit the cell is completely inert.

Successive stages in design have made possible a considerable increase in plate area by reducing both the plate thickness and the separation. The adoption of a narrower plate pocket and several minor alterations and adjustments made to terminal connectors, etc., together with improvements in active material, have all contributed towards reducing the internal resistance and improving performance so that a high voltage can now be maintained under the heavy discharge required for battery manœuvring or starting in the case of a C.I. engine. A considerable saving in weight and overall dimensions has also resulted. No alteration in regulator setting is required from that suitable for a lead acid battery.



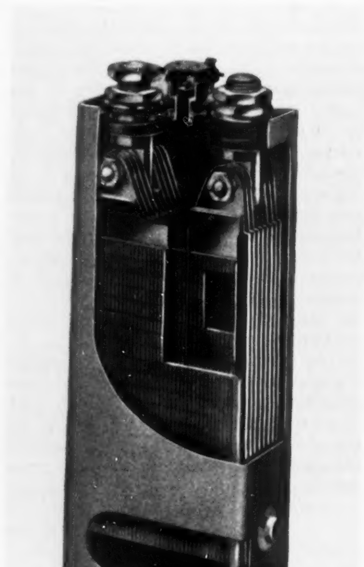
Alkaline battery—former design

The positive and negative plates are identical in mechanical construction. Active material is pressed into briquettes before being fed between two finely perforated strips of high grade steel which are sealed along their edges to form tubes. These are cut to length and mounted in a welded steel frame to form the plate. The active material is thus enclosed in finely perforated steel envelopes so that there can be no sediment formed in use.

The active material of the negative plate is a preparation of cadmium oxide with a small quantity of iron oxide. The positive plates consist mainly of specially prepared nickel hydroxide mixed with other ingredients to increase the conductivity and porosity of the mass. The electrolyte is mainly a solution of potassium hydroxide in distilled water and has a density of 1·23 to 1·16. This does not vary with the state of the charge as it does not enter into chemical action.

On discharge, the nickel oxide in the positive plate is reduced to a lower form while the cadmium and iron in the negative plate are oxidized to cadmium and iron oxides respectively. On charge the reverse action takes place, the metallic oxides of cadmium and iron being reduced to the pure metals and the lower nickel hydroxide restored to its original condition.

The characteristics of an alkaline battery in use differ in many ways from that of a lead acid battery of about the same capacity. For instance, it is well known that the capacity of a lead battery falls gradually during its life, which probably will not exceed four years under normal conditions. The alkaline type usually shows an increase



Alkaline battery—current design

in capacity during the first year or so and should it decrease appreciably later, it can generally be brought back again by a charge of electrolyte and hard cycling, the ultimate life being about equal to that of the chassis.

The two types behave differently under heavy discharge. For instance, a lead battery will not give its ten-hour rated capacity over shorter periods, the capacity falling as the current is increased. An alkaline battery of the same rating maintains its capacity up to high rates of discharge, though the lead battery can give a momentary higher output. This ability to maintain capacity is of value in certain heavy duty installations as it obviates the need for a large battery to ensure satisfactory service.

Under working conditions when discharge occurs immediately after charge, the initial portion of the discharge curve is at a substantially higher voltage than the normal and it takes an appreciable time to fall to the normal value. Conversely, the initial portion of the charge curve immediately after discharge is at a lower voltage and it takes a little time to adjust itself to the normal figure. In consequence, the transition between charge and discharge is a smooth curve rather than a sudden change, and the general effect is very much more evident than in the case of a lead acid battery.

AUTOMATIC CHASSIS LUBRICATION

A Clayton Dewandre Development for Trailers

BECAUSE of the economies in vehicle operating costs, time, labour and lubricant that result when automatic chassis lubrication is employed, there has been an increasing demand for Clayton Dewandre lubricators for all types of vehicles. For long-distance service vehicles, the belt driven "Multipoint" lubricator, mileage controlled, is made in various sizes capable of feeding from 18 up to 72 separate chassis points. To meet the needs of passenger vehicles engaged on city or urban services and goods vehicles operating on short delivery runs, where the brakes are used fairly frequently, the Automatic lubricator was developed. This unit is actuated by a small pneumatic cylinder, connected into the brake line of the vacuum or air-pressure brake system, so that the lubricator is operated every time the brake pedal is depressed.

Heavy trailers are invariably used in conjunction with long-distance vehicles and therefore lubrication should be on a mileage basis. The "Multipoint" lubricator cannot be used for such an application since it requires a belt drive, and although it would be a simple matter to fit an automatic machine operated with the trailer brakes, the infrequency of brake application would prevent efficient chassis lubrication.

One solution to this problem is to fit a "Multipoint" lubricant to the prime mover vehicle, which lubricates the chassis and at the same time actuates a distributor valve to operate an Automatic lubricator on the trailer. The valve is built into the blank end cover of either a 24 or 36 point machine. It is supplied from the vacuum or air-pressure brake reservoir on the tractor vehicle and is connected to the trailer lubricator through a flexible hose and coupling.

A second revolving face cam in the "Multipoint" lubricator causes the distributor valve to open to permit vacuum, or air pressure, according to the system, to be communicated to the trailer lubricator which is then released. This action produces impulses which operate the trailer lubricator at the same feed rate as the tractor "Multipoint" lubricator.

One of the first installations of the new equipment has recently been completed by Scammel Lorries Ltd. on an eight-wheel articulated lorry for the Ever Ready Co. (Great Britain) Ltd. The vehicle operates between London and Wolverhampton and covers 700 to 800 miles per week. There is no doubt that the automatic chassis lubrication will not only reduce wear at the chassis bearings but will also cut maintenance time and thereby improve delivery schedules.

On the tractor chassis a 36-point "Multipoint" lubricator feeds 38 tractor chassis bearings; four of the less important points have split feeds. The points lubricated are:—

Front and rear spring shackles and pins ..	12 points
Steering drag link and track rod	4 points
King pins	4 points
Brake, clutch and accelerator linkage bearings ..	12 points
Turntable	4 points

The semi-trailer is equipped with an air-pressure operated 24-point Automatic lubricator that feeds 19 separate chassis bearings; four points have double feeds. The points lubricated are:—

1st and 2nd axle fixed spring pins ..	4 double feed points
1st and 2nd axle shackle pins	8 points
Balance beam centre bearing	2 points
Brake gear	5 points

The remaining feed point on the lubricator is used to lubricate the air pressure Servo piston of the lubricator.

PLASTICS MOTOR BODY

A GLASS fibre-polyester two-door saloon body for mounting on a standard Lancia chassis is being shown at the British Plastics Exhibition which is now open at Olympia, W.14. A noteworthy feature of the design is that the body can be removed from the chassis in a few minutes simply by undoing eight bolts, unplugging the electrical connections, and lifting it off. The chassis is then left in a driveable state.

Although the body shell, which is made in one piece, is strengthened with $\frac{3}{8}$ in diameter by 16 s.w.g., cold drawn, mild steel tube, it weighs only about 110 lb. The tubes were made to conform with the body lines and all important joints were reinforced with welded-on mild steel gussets. Before the laminations were laid on, the tubes were bound with glass cloth, and the body frame was placed on a plaster of paris, male mould. The laminations were bonded to the tubes which were totally enclosed by laying strips of glass cloth over them on the inside of the shell and applying the resin after removal from the mould.

The body is shown in a partly finished state, and is not completely rubbed down and prepared for cellulose When finished, it will incorporate glass cloth-polyester, top hat section stiffeners in the wings and body sides. The doors will be stiffened with similar top hat sections and with low density material. All window channels will be of laminated plastics instead of the more usual brass channel sections. A fully ducted radiator is incorporated, and the head lamps are mounted in the duct. Flush fitting, pull-out type door handles are to be used and the radio aerial is moulded between the laminations in the roof. All these features are designed to give clean exterior lines.

Apart from this body, there are other exhibits that may interest automobile engineers. It is surprising what a large variety of finishes are available on plastics panels, and a number of mechanical components are now made of laminated and moulded plastics. This is a relatively new industry that offers enormous scope for further development. (2047)

A NEW COATING ALLOY

A Development for High Temperature Use

THE use of leaded fuel and high boost in aircraft I.C. engines imposes very severe conditions on the poppet valves. Present day flight schedules demand a maximum period between overhaul and complete reliability to avoid unscheduled stops. To a lesser extent this applies also to modern engines in land vehicles where efficiency and hard running are demanded.

At present, the various alloys used for valves, valve seat faces, or completely coated heads, either fail from lead oxybromide attack when the valve temperature exceeds a critical figure, or from mechanical weakness at these elevated temperatures, or a combination of both.

Messrs. Rolls-Royce Limited have, during the past few years, developed and introduced into flying service a new coating alloy which is now being used regularly on valves in civil aircraft engines. The use of a coating of this alloy on the valve head and seating has increased the period between scheduled valve overhauls to 1,000 hours and even after this time only a small proportion of valves are withdrawn from service.

The new alloy, known as C.26 (covered by Letters Patent, throughout the world), has a nickel base containing chromium, aluminium, molybdenum and other minor additions in a carefully balanced composition based on an investigation of the alloy system of the four major constituents. The manufacturing and marketing rights have been acquired by Deloro Stellite Limited, Birmingham. Although hard, the alloy is not brittle under the repeated impact given to a valve face and does not crack radially as many exhaust valve coats do under the time/temperature stresses imposed by high power engines.

Although the temperatures of inlet valves are somewhat lower, the new alloy has been adopted on them owing to its excellent resistance to the abrasive action of the products of combustion of leaded fuel.

The better wearing characteristics of C.26 result in a very low loss of tappet clearance. This is an immediate economy in terminal maintenance; a considerable item when it is remembered that a large passenger plane has 192 valves to be checked and readjusted when necessary.

Laboratory, engine and service tests have shown the superiority of C.26 alloy on aircraft valves and other uses are being explored in view of its excellent oxidation resistance at all temperatures highly resistant to lead oxide and lead oxybromide at temperatures up to 800 deg C. and above this temperature it is not safe to run modern highly stressed exhaust valves for long periods. The alloy is resistant to molten caustic soda and has been used in the form of cast containers for fluxes containing fluorides.

C.26 is deposited on to I.C. engine valves by means of the oxy-acetylene torch. A flux may be used if preferred, but no intermixing with the base metal must take place. Where the iron content is not critical the arc welding process is used, and casting by normal commercial methods does not present any difficulties. The alloy has a fine structure and fracture and is machinable without undue difficulty. Some up to 1,250 deg C. (the drop in pressure measured after six hours in a closed system containing C.26 and oxygen or air at this temperature is negligible). Provided that the iron content of the alloy is kept below 1·0 per cent, it is properties of the alloy are listed above.

C.26 ALLOY

Melting range	1,280-1,360 deg C
Density (cast bar)	7·5 to 7·6 gms/c.c. at 20 deg C
Mean coefficient of thermal expansion (20-800 deg C)	14·9 to 15·2 $\times 10^{-6}$ deg C
Thermal conductivity (800 deg C)	3·2/3·4 $\times 10^{-4}$ CHU/in/sec
Ultimate tensile strength (aged)	
20 deg C	60-70 tons/in²
800 deg C	42-48 tons/in²
Hardness D.P.N. (30 Kgs)	
At 20 deg C As cast	390
Aged 16 hrs at 700 deg C	490–525
Aged 16 hrs at 800 deg C	400–440
At 600 deg C	420
At 700 deg C	395
At 800 deg C	350
Alternating bending test in contact with lead oxybromide	
at 700 deg C	$\pm$12·8 tons/in² safe stress for 40 $\times 10^{6}$ reversals

The hardness figures quoted for 600, 700 and 800 deg C, are the average of five impressions, each of 15 sec loading obtained with tungsten carbide using a new pyramid for each test. (2046)

PISTON RINGS

IN an article entitled "Piston Rings for High Performance Engines," by E. W. Portmann published in German in *M.T.Z.*, September 1952, the application of formulæ for the elastic modulus to self-tightening rings is explained. Such rings have a higher contact pressure at the gap and opposite the gap. Loading, or the closing force, is relatively strongly affected by the elastic modulus and ring gap. To meet the requirements of higher compressions and/or higher engine speeds, the elastic modulus currently adopted in the United States is 12,000-14,000 kg/mm². Values of ring-gap equal to 12-18 per cent of the ring diameter are usually enough to balance the variations of elastic modulus. The equation for tension is given, and the difficulty of experimentally determining reliable tension values is explained.

Non self-tightening rings may be mechanically formed by hammering on the inner side with a special machine, or thermally formed by controlled heat treatment of the ring while it is in contact with a precision former. Self-tightening rings are cast from a pattern that is not truly circular in shape. Various stages in the manufacture of both non-round and profile-turned types are explained by reference to diagrams.

Recommendations concerning rings for use on high performance engines are: (1) Self-tightening rings, preferably of the profile-turned type, are more effective because of the variable pressure distribution. (2) Chromium plated top rings, shouldered rings being better than rings with the normal right-angled cross section. (3) The axial width is all-important for engines turning at high rotational speeds. Tests in which thickness of wall, tension, ring cross-section and roughness of the working surfaces were kept constant and with rings of different axial width, revealed the "critical ring width." (4) By increasing the wall thickness a to D/a=20, vibration is made less likely because of the higher natural frequency. (5) The running-in process is facilitated by as fine a finish as possible of the cylinder bore and by a suitable degree of roughness of the ring surface. Shouldered rings run in more easily than normal rings. Heavy duty oils are not suitable media for lubrication during the running-in period. *M.I.R.A. Abstract* 6190.

WELDING DEVELOPMENTS

A Review of the Methods Employed for Vauxhall Velox and Wyvern Bodies

WHEN Vauxhall Motors Ltd., Luton, decided to introduce two new passenger cars, the Velox and the Wyvern, there was a necessity to develop completely new body building equipment. The change-over to these new models occurred during a period of extensive factory reorganization and the opportunity was taken completely to reorganize the whole of the passenger car body building lines. For certain functions, techniques which were completely new to this country and which, to the best of our knowledge, are still peculiar to the Vauxhall organization, were adopted.

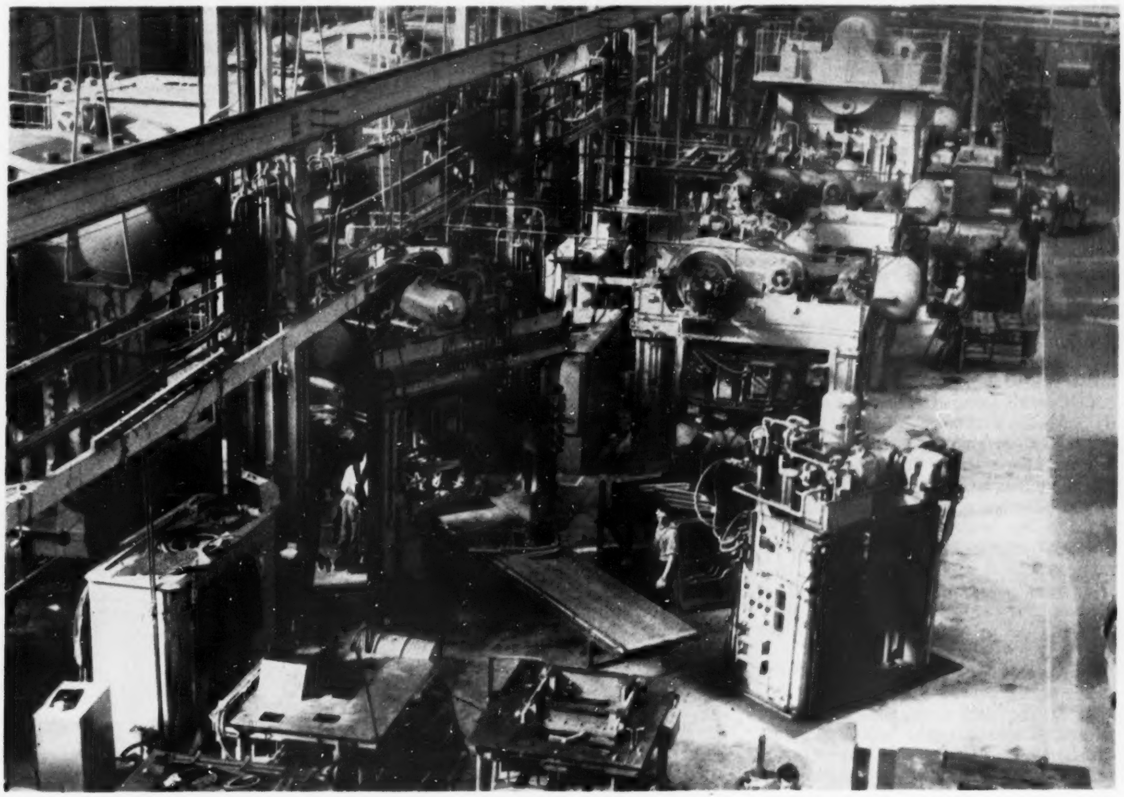
The problems to be faced were eased by the fact that the body design is exactly the same for both models, but certain inescapable factors introduced production problems of some complexity. There were two main causes for these production problems ; first, the need for a balanced production correlated with the planned output of vehicles; and second, the fact that the body building functions had to be installed in existing factory buildings. Since some of the techniques that were to be introduced were such as to give an output much in excess of the planned maximum, it became necessary to use certain equipment for more than one component in order that there should be economic machine utilization. This, of course, entailed batch production, which in its turn raised the problem of work storage.

As the body building processes had to be carried out in existing factory buildings, it was impossible to employ the ideal lay-out in which the body-building lines would be a continuation of the press shop. In fact, the body building is carried out in a section that is separate and at a different level from the press shop. Because of this it was necessary to plan for relatively lengthy transport of materials from the press shop to the body building section. This phase of production was the subject of detailed analysis, concerned not only with the actual handling, but also with the storage of components that batch production necessitated. The complete lay-out for both materials handling and actual production functions is one of which any organization could be proud.

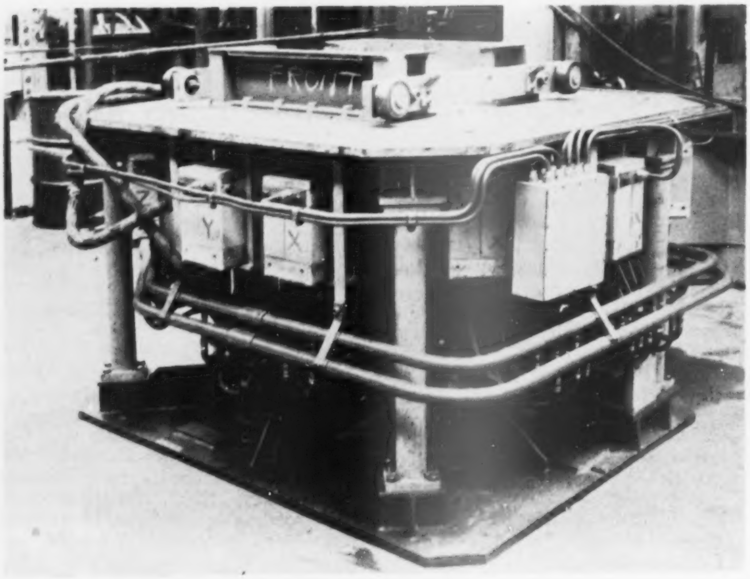
Press welding

Certainly the most spectacular innovation was the introduction of press, or Hydromatic, welding, for suitable applications ; this development for fabricating certain sub-assemblies will be discussed before the body building section is described. There are two press welding sections, one for door and luggage lid assemblies and the other for front and rear underbody, sill panel and seat frame assemblies. The first of these will be described in some detail.

Although these notes are primarily intended to describe the methods used for fabricating major sub-assemblies and complete bodies, reference must first be made to the manner in which the press section and the Hydromatic welding sections for doors and luggage lids have been integrated. It has been possible to install the welding equipment



The press welding section at Vauxhall Motors Ltd. for doors and luggage trunk lids for Velox and Wyvern cars



Top and bottom halves of press welding unit

immediately adjacent to the presses for the major components and there is continuous flow production through the various press operations and then through the welding section until the sub-assembly is fabricated and ready for transfer to the body building section.

This Hydromatic welding section is used for the production of five different sub-assemblies, front and rear near and off-side doors and luggage trunk lids. Adjacent to the welding section there is a line of eight Clearing presses on which the main components are produced. Analysis has shown that the optimum economy of output is obtained by running sequentially for two days on each sub-assembly.

A brief description of the press sequence for front doors will suffice to illustrate the methods employed. Outer and lower inner door panels are produced in pairs on five presses. Each press has two sets of tools mounted on a sub-plate. It is possible to assemble doors at the rate of 300 per hour and to maintain a continuous flow of pressings to the welding section it is necessary to make effective use of some 65 per cent of the possible press strokes. Press loading and unloading times and the time for transfer from one press to the next had to be very carefully considered if this rate of press utilization was to be maintained. Loading is effected manually, but ejection is automatic and of such a character that the work is ejected direct on to a mobile power band or slat conveyor for transfer to the loading station for the next press in the line.

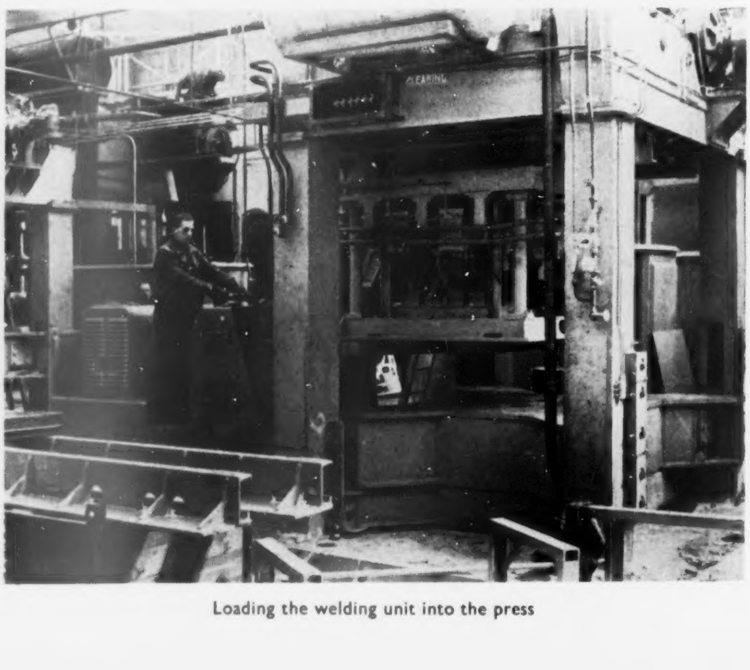
The press sequence for the outer and lower inner panels for a front door is that at the first press the panels are drawn to form ; at the second they are trimmed ; at the third the outside flange and the flange of the window opening are formed on the outer panel while the inner lower panel is restruck; at the fourth, the push button hole in the outer panel and holes in the side walls of the lower inner panel are pierced and the window opening flange in this panel is formed ; at the fifth and final press operation the window opening rebate of the outer panel is formed while several holes are pierced in the lower inner panel.

Various means of automatic ejection are employed on these five presses. For example two Sahlin iron hands are fitted to the first press in the line. As the press opens these devices come into action to grip and lift the panels by means of pneumatically-operated fingers. As the upward stroke of the press continues, the iron hands swing out and at the appropriate point the fingers release and the panels drop on to the power conveyor.

Other presses have different types of ejectors. One method is to have slide rails which rest below the die face during the working stroke, but as the press opens the rails are raised by a small air cylinder to form a ramp down which the panel slides to the conveyor. The rails lift the panel clear of the die, and at the top position trip a valve to actuate an air cylinder that starts the panel to move down the ramp. In other cases the ejection device within the die has composition rollers that rise to become in effect a short gravity roller conveyor for carrying the component from the press to the inter-press conveyor.

From the fifth press, the outer and lower inner panels pass to the press weld section by different routes. The lower panel passes through the press on to a conveyor that takes it to the first press welding machine. Meanwhile, upper inner panels are produced on the three remaining presses in the press line with work flow in the opposite direction so that upper and lower inner panels come together at the end of the press operations.

Although only three presses are used for producing the upper inner panel, there are five operations. This is effected by having two sets of tools on a sub-plate in each of two presses. At each of these presses the work is, of course, first loaded into one set of tools and then transferred to the other set for the next stroke. The press sequence is :—blank and pierce from sheet in the first press ; form embossings at the first stroke of the second press then transfer to the other tool for trimming the sides and piercing holes ; at the third press the flanges are formed in the first tool and restruck to correct shape in the second.



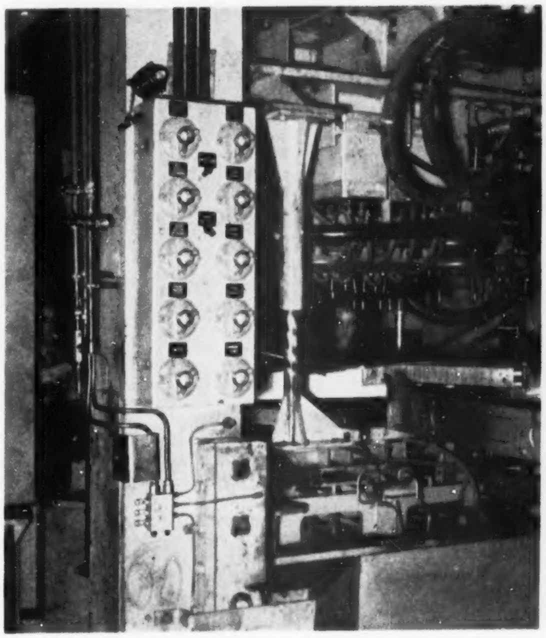
Loading the welding unit into the press

Press welding machines

There are eight press welding machines in the section for doors and luggage trunk lids. They are of two types, one a 30-ton four-pillar type known as the EP8, and the other an open-face type known as the EP24. They differ considerably in size, construction and welding capacity, but generally employ the same principles, and therefore only the EP8 type will be described in detail.

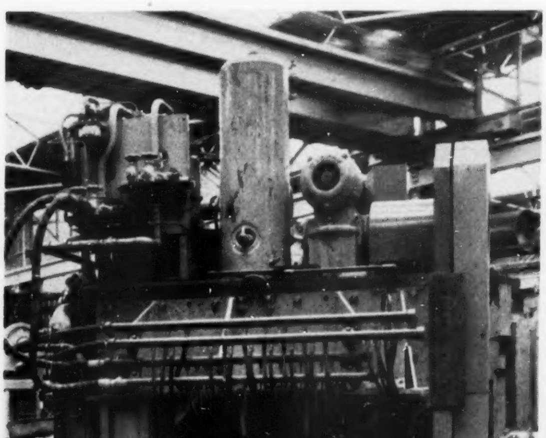
The framework of the EP8 machine is a British Clearing press supplied by the Rockwell Machine Tool Co. Ltd., to Vauxhall design. Essentially, the structure comprises a fabricated steel crown supported on four hollow pillars ; each pillar has machined faces on the lower part to form ways for the up-stroking table. A 10 h.p. motor mounted on the machine crown supplies the power to a two-throw crankshaft through an air-operated plate clutch controlled by a solenoid. Rotation of the crankshaft imparts an oscillating motion to countershafts on either side of the crown, and through a system of levers the oscillating motion effects the vertical motion of the table. There is a regular pattern of holes in the crown face and the moving table for attaching the upper and lower halves of the welding fixture to the machine.

For supplying pressure to the welding guns, the machine has a self-contained hydraulic unit. Except for a group of cam switches that are operated by the main driving shaft, all the controls are housed in a steel cubicle at the side of the machine. These controls include an auto-transformer heat control switches, Ignitron contactors and a welding timer for operating the welding sequence. Provision is also made for independent control of the various machine functions.



The control panel on a press welding machine

Essentially, the fixtures used in a welding press are comprised of upper electrodes attached to the crown of the press and lower electrodes mounted on the table. The upper half of the fixture has a fabricated steel framework on which are mounted the hydraulic cylinders that provide the welding pressure, the transformers for supplying the welding current, an oil manifold for supplying oil under pressure to the welding cylinders and a water manifold for supplying cooling water to the electrode tips and the transformer secondaries. The lower half of the fixture has a main base for carrying the backing electrodes and locations for holding the components in correct relationship during the welding operation.



Set-up for welding the hinge reinforcements to the door inner panel

The welding guns are of Vauxhall design and manufacture. Each gun is an individual unit comprising a double-action hydraulic cylinder with an electrode holder and tip attached to the piston rod. Protective insulation prevents shorting between guns in close proximity to each other. Two sizes of hydraulic cylinder, $1\frac{1}{4}$ in and $1\frac{3}{4}$ in diameter bore, are used. The larger size is used for welding the heavier gauge materials or to provide the greater pressure needed for direct welding. As far as possible welding is carried out with the electrodes vertically disposed, or only at a small angle from the vertical. This disposition of electrodes is desirable but not always possible. Where it is necessary to make welds in a plane at right angles to the table surface, pincer guns are used.

Three spot-welding techniques are employed. The most widely used is series welding in which two electrode tips on one side of the work are connected in series and the current is shunted across a backing electrode on the other side of the work. This uses a little higher current than is needed for simple spot welding, but two spot welds are made, one under each tip. Where it is important to avoid weld marks, indirect-series welding is employed. For this, backing electrodes insulated from each other are used beneath each electrode tip, and the current is shunted through the work instead of through the backing electrode. When welds are to be made near to the edge of a panel of heavier gauge material, direct welding is used. This effects only one spot weld since the current is returned from the backing electrode through a knife switch that connects the top and bottom halves of the fixture when they are closed together.

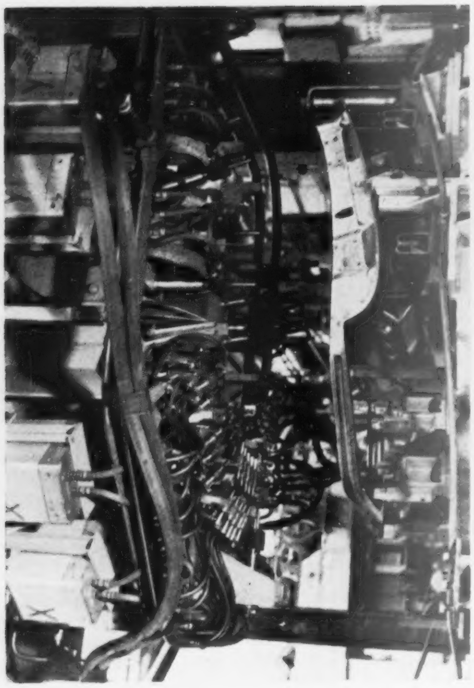
The cycle sequence for a press welding machine is :—

(1) Depression of a push button energizes the motor control circuit and the table is raised to the welding position.

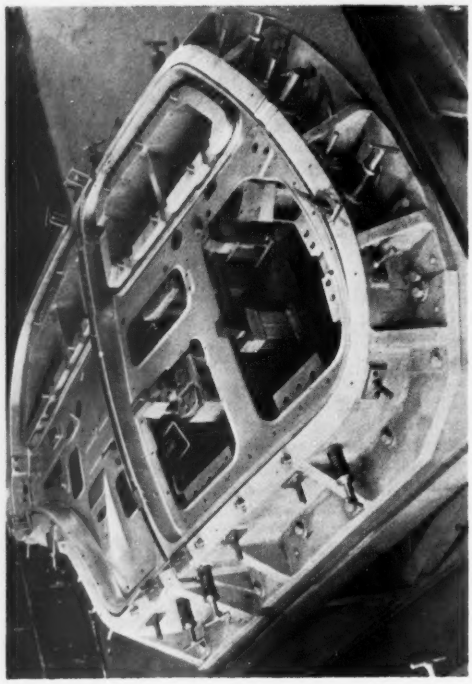
(2) When the table reaches the welding position, the hydraulic controls are actuated and welding pressure is applied to the welding guns.

(3) Immediately the welding pressure is established the welding timer fires the Ignitron contactors in progressive sequence.

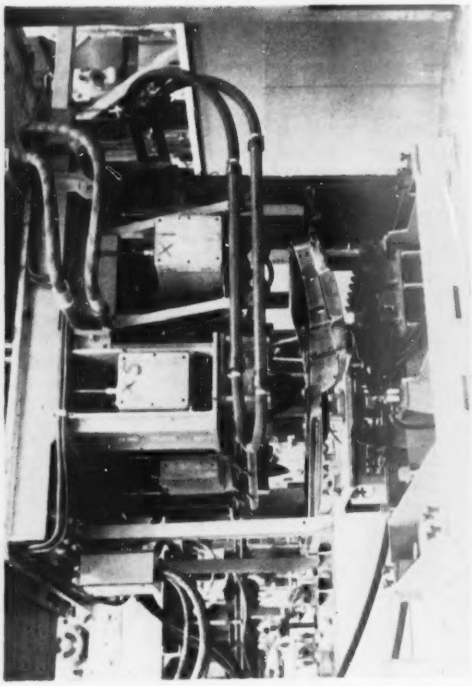
(4) When the last contactor has fired,



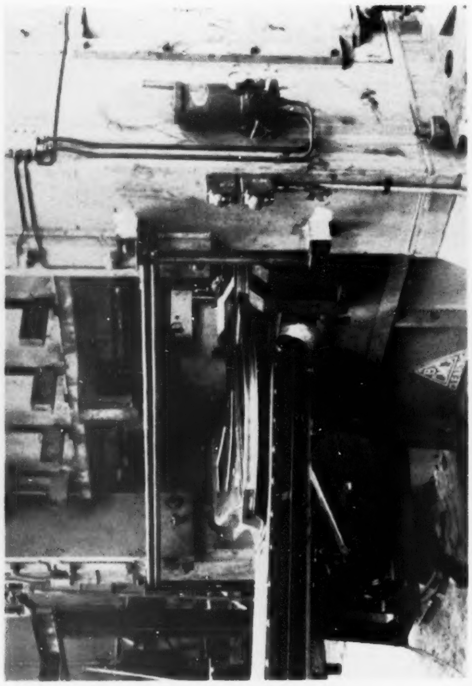
Welding the inner panel to the outer door panel



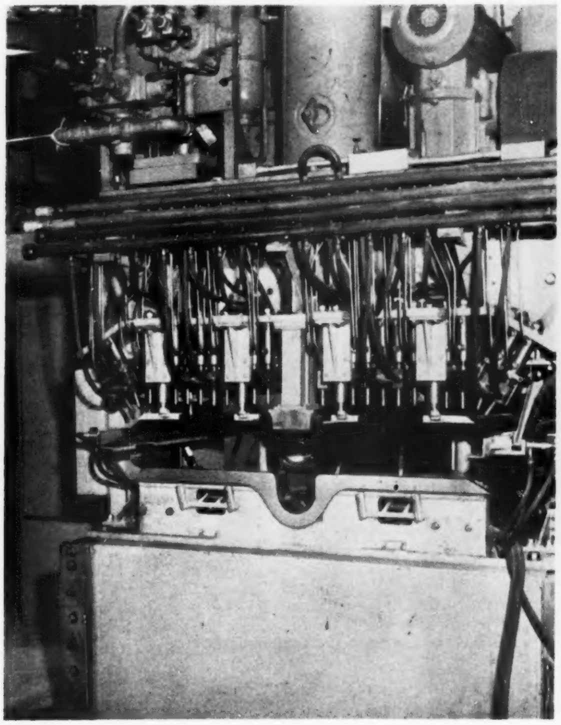
Acceptance fixture for completed door assemblies



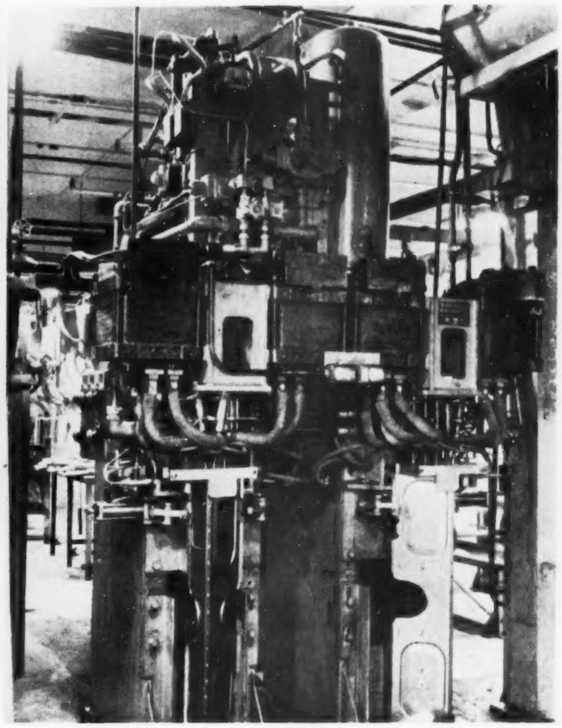
Welding the door pocket to the inner panel



Door assembly passing into the hemming press



Set-up for welding front underbody floor panel to tunnel reinforcement



Welding the end plates to the front seat base in the underbody press weld section

hydraulic controls release the pressure from the welding guns and the electrodes are retracted.

(5) The table is then lowered to the open position and the work is ejected from the press.

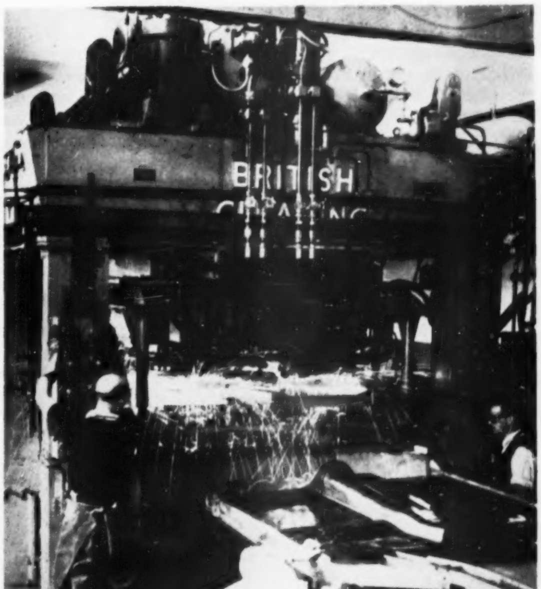
The upper and lower halves of the welding fixture are independent units and it is essential that they are correctly aligned in relation to each other when in the press. For this reason set-up posts are incorporated in the fixture design. EP8 type fixtures have four posts and EP24 fixtures two posts. These set-up posts afford a convenient means of keeping the halves together when a fixture is being handled or stored.

One of the most important factors leading to the decision to install these expensive press welding machines was that the quality of the welds could be much higher than could be obtained by more conventional methods. To maintain this quality it is essential that the electrodes are always in good condition. Special portable pneumatic tools are used to dress the electrode tips at definite intervals. They are adapted right-angle nut runners that can be connected to the air supply on the press. A specially designed cutter is fitted in the tool, and the electrodes can be dressed without being removed from the press. A backing electrode consists of a water-cooled base to which replaceable blocks of the appropriate form are attached. These can readily be adjusted to compensate for wear.

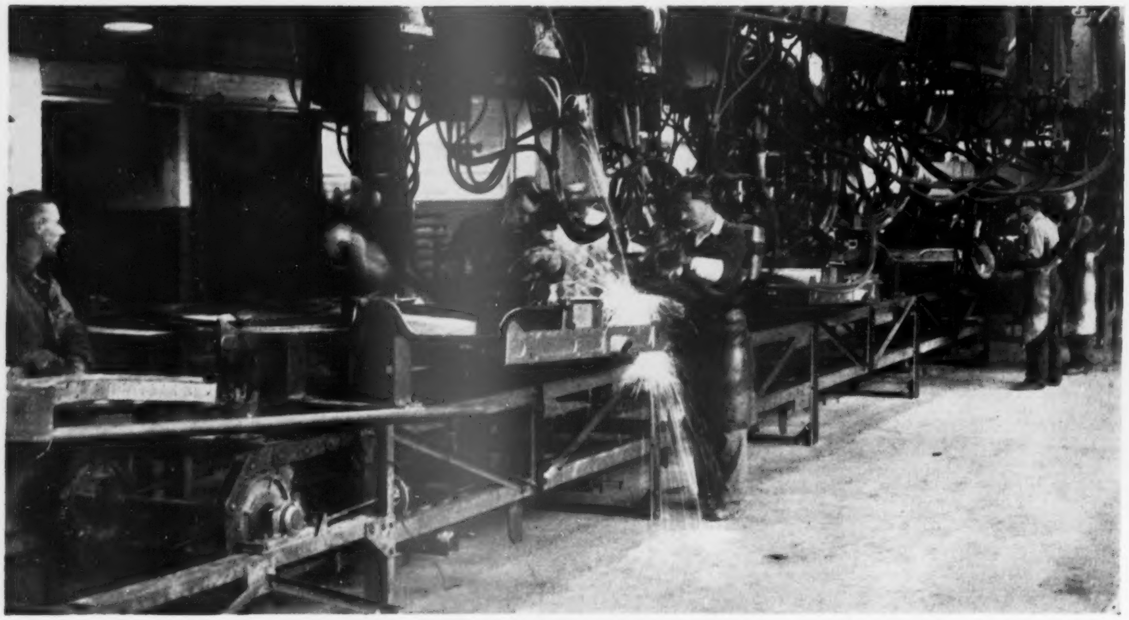
As the planned production for this section calls for consecutive runs of two days each on the five assemblies, provision has been made to allow accurate change-over to be made as quickly as possible. To begin with the set-up posts allow the top and bottom halves of the fixture to be handled as a single unit. This unit is usually lifted into position by a 5-ton Baker fork-lift truck. Rollers on top of the fixture and runways on the press platen allow the fixture to be slid into position. The necessary oil and water connections are made by means of Lockheed quick-release couplings and there is a terminal box for electrical connections.

Door fabrication

As was stated earlier, the main



A press weld machine during the welding cycle on a front underbody



Carrying out addition portable gun welding on the final conveyor for front and rear underbodies

components for the door assembly, the outer panel and the upper and lower inner panels are produced on presses immediately adjacent to the press-welding section to which they are conveyed on mobile power conveyors. Other subsidiary components are produced in another section of the press shop and are trucked to the press-weld section in stillages.

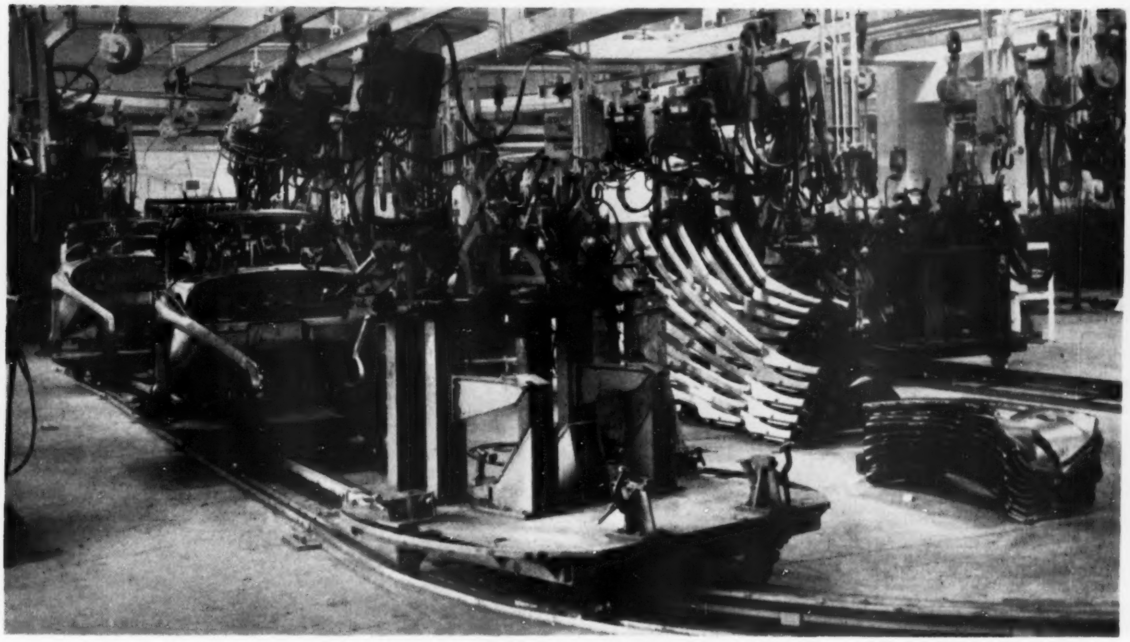
The first press welding operations are carried out on the upper and lower inner panels. To begin with the inner upper panel is placed in the lower welding fixture together with an upper corner patch and a hinge reinforcement. Meanwhile a window buffer bracket is fitted to the inner lower panel while it is being transferred on a mobile power conveyor from the final press operation to the first press welder. This bracket is self-locating. At this first welding operation 21 spot welds are made.

When the first welding operation is completed the assembly is ejected from the fixture by an air-operated ram and pushed on to the conveyor for transfer to the next machine. At the second machine, an EP24, 14 spot welds are made to seal the hinge assembly. The inner panel assembly is completed at the third machine. On this machine 26 welds are made, some for welding in the pocket panel and others as strengthening welds for the flange. At this stage the inner panel assembly is ready for fabrication to the outer panel assembly.

After the final press operation on the outer panel two welding operations are carried out before the panel reaches the press welding line. These operations are carried out individually because the welds are in positions that could not conveniently be reached on a press-welding machine. At the first a spacer is welded in by means of a portable welding gun at a station beside the press line; at the second the door handle reinforcement is welded in on a 300 kVA projection welder. After the outer panel leaves the projection welder and before it reaches the assembly line, a Bostick deadener pad is rolled on the inner face.

The outer and inner panels meet at a clinching fixture. This fixture is pneumatically operated and electrically controlled. The outer panel is first located in the fixture, and the inner panel assembly is fitted into it so that the outer panel can be lightly clinched

Special Vauxhall portable welding guns. The electrode holders in the gun on the right can be swivelled through a complete circle



Front end assembly conveyor track

over the inner panel flange to hold the two panels together until they are welded.

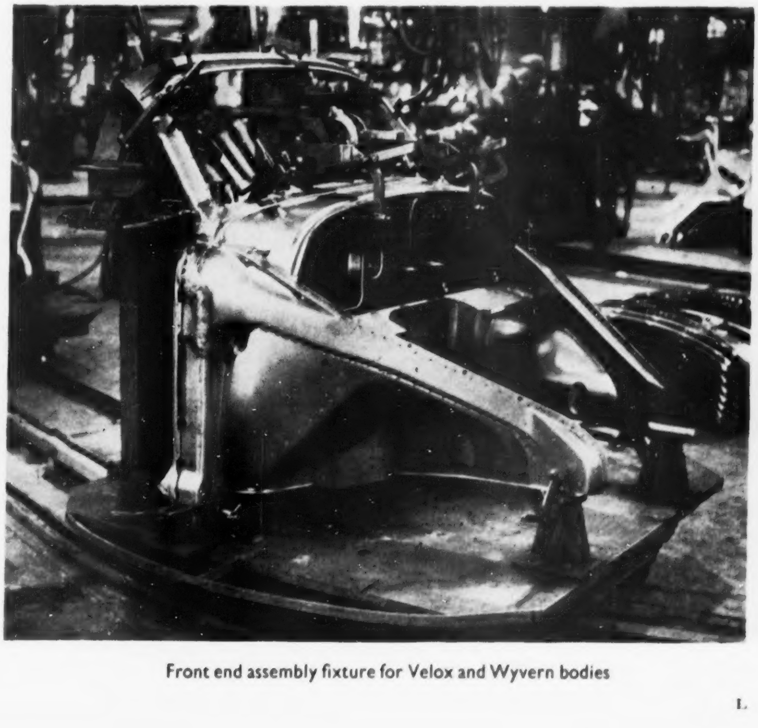
From the clinching fixture the assembly is conveyed to an EP8 press welder where welding is effected round the flanges and the window aperture. From this machine the assembly is ejected on to a conveyor for transfer to a hemming press. Suspended above this conveyor is a portable spot welding gun for welding the door panel spacer to the inner panel at five spots. A conveniently placed floor switch allows the operator to stop the conveyor travel while the welding is carried out.

The door hem is turned and flattened in the hemming press. An air-operated roller ejector, with composition rollers to prevent damage to the surface of the door, is incorporated in the hemming die. After the door leaves the hemming press, the assembly is completed on a welding press that is tooled to weld the hem in 32 places. Complete door assemblies are hung in pairs on an overhead conveyor for transfer to a storage area and subsequent issue to the body building section.

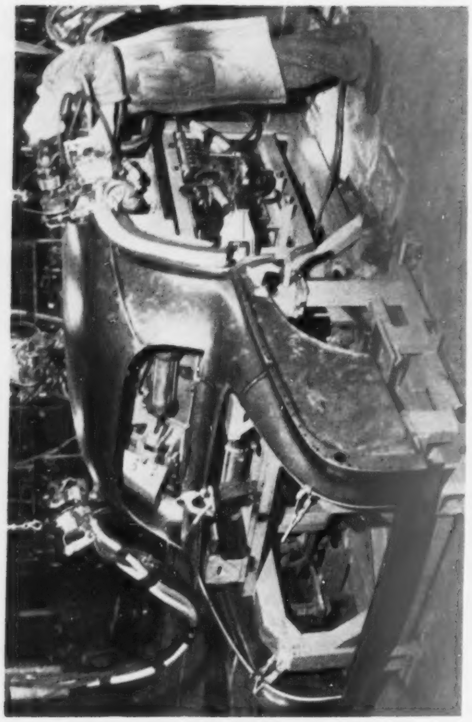
Press welding can be carried out efficiently only if the press work is carried out to a high standard of accuracy, since the output can be maintained only if the various pressings can easily be located in the fixtures and as easily removed when welding is completed. It is also important that the finished sub-assembly should be accurate in order that the output can be maintained in the body building section. To ensure that this accuracy is maintained, sub-assemblies are periodically checked in an acceptance fixture.

So far as the quality of the work is concerned, press welding is undoubtedly far superior to the more conventional methods. It also has considerable economic advantages. For example, to maintain a supply of doors to meet an output of 300 cars a day, the methods previously used called for 54 operators and an assembly area of 9,320 square feet. For fabrication by press welding the assembly area occupies only 3,640 square feet and 24 operators are capable of producing 300 doors per hour. There is also a further economy in that because of the better quality and greater consistency of the welds, metal finishing operations to remove welding blemishes from panel faces have virtually been eliminated.

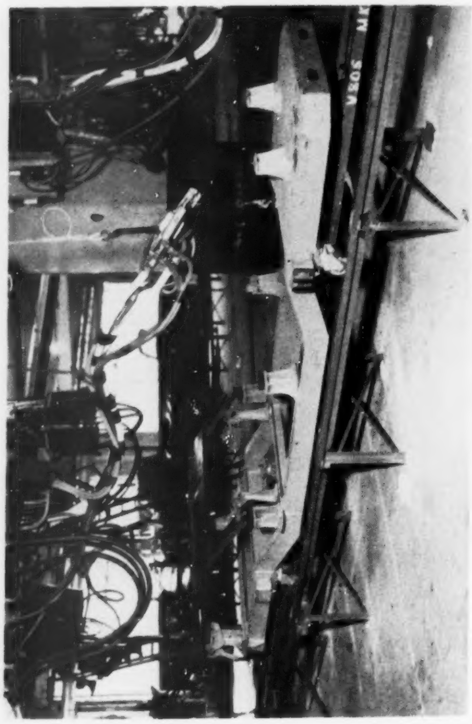
Press welding is also carried out in the fabrication of front and rear underbodies, sill panel assemblies and seat frame assemblies. All these assemblies are produced in one section. The press-weld machines are of the same types as those already described. In this section, however, all the press weld machines are inter-connected in



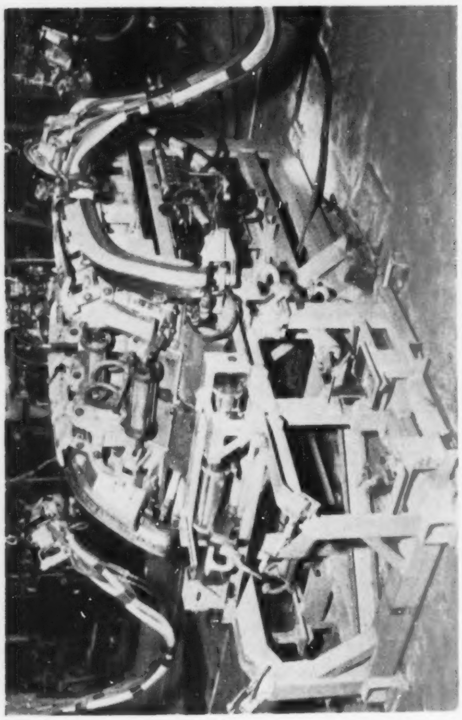
Front end assembly fixture for Velox and Wyvern bodies



Welding roof channel to roof panel side and rear deck

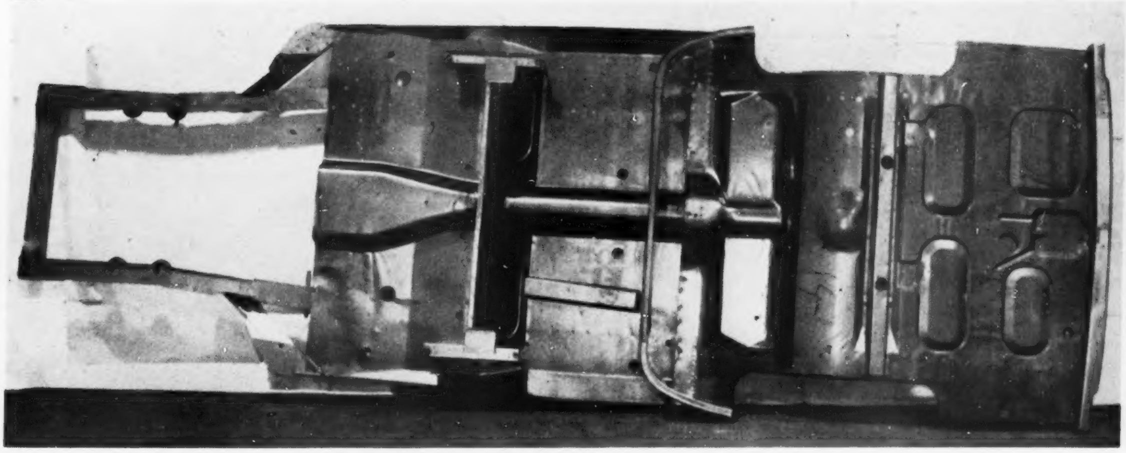


Bogie for carrying the front and rear underbodies



Fixture for assembly of roof channel to roof panel side and rear deck

Fixture for welding the roof panel to the rear quarters



A completed underbody

such a manner that not more than one machine can be fired at a time. The reason for this is that the current demands for certain machines are so high that if two or more were fired at once, there would be a severe voltage drop with a consequential detrimental effect on the efficiency of the welding.

For technical reasons it is not practical to complete the fabrication of, say, a front underbody on the press welding machines. Therefore, when the underbody leaves the final press welder it is pushed along rails on to a raised conveyor above which portable welding guns are suspended for use in positions that could not be reached satisfactorily on a press welding machine or for making additional spot welds where the specified pitch of the welds is too close to allow all the welds to be made on the machine. Complete front and rear underbodies are delivered separately, but in pairs, to the body building section.

The body building section

The body building department may be regarded as consisting of five sections, namely :—

(1) Front end sub-assembly section.
(2) Front end assembly section.
(3) Rear end sub-assembly section.
(4) Rear end and roof assembly section.
(5) Final assembly section.

Each sub-assembly section is conveniently situated in relation to the section which it is to feed so that work transport is kept to a minimum. In general, the sub-assemblies are welded on static machines by conventional methods that do not call for comment.

Each front end assembly is fabricated on a single fixture. In all there are 15 fixtures each mounted on a bogie that is carried round an enclosed loop floor conveyor. Welding guns of the appropriate types for each of the welding operations carried out during the circuit round the track are suspended above and to one side of the conveyor. Incidentally, the welding guns in this and all other welding sections are of Vauxhall design and make. This practice has obvious advantages. Every gun is specifically designed for the operation on which it is to be used and consequently optimum operational efficiency is obtained. Furthermore, the maximum degree of standardization in equipment has been achieved with the result that maintenance is simplified and any necessary repairs or replacements can be effected easily.

The layout is so planned that a definite assembly sequence must be followed, and to suit this the various component sub-assemblies for the front end assembly are delivered as conveniently as possible to the locations at which they will be required. Substantially, the front end assembly sequence is :—

(1) Locate dash panels in fixture.
(2) Assemble demister air duct to top dash panel.
(3) Locate dash side panels and front pillar assemblies.
(4) Locate scuttle panel in fixture.
(5) Locate instrument panel rail assembly.
(6) Fit upper dash side panel.
(7) Fit wind-screen top rail.

As can be seen from the illustration of one of the front assembly fixtures, particular care has been exercised in the design to allow easy access to all the points to be welded. It will also be noted that toggle clamps are widely used to allow the work to be fastened securely in the fixture with a minimum of effort and time. Considerable use is made of location holes, pierced during the press operations.

Although by far the greatest amount of welding on the front assembly is carried out by means of portable guns, provision is also made for gas welding where exposed joints need to be filled.

A portable seam welder in use

At the completion of the fabrication, the assembled front end is removed from the fixture for transfer to the main body building where it and the roof and rear quarters assembly will be mounted on the underbody ready for the fabrication of the complete body shell.

Assembly of the roof and rear quarter is carried out in a series of static fixtures. The complete assembly is effected in four fixtures. In one fixture the cant rail roof band and rear diagonal member are welded together, while at the same time the roof panel is welded to the rear quarters sub-assembly in a second fixture. These two assemblies are then mounted in a third fixture and welded together. For the final operation, the work is transferred to a fourth fixture where the centre pillar and sills are welded in. This fixture is designed to ensure that the correct pillar position is maintained. There are eight fixtures in all to produce the required output, three fixtures for each part assembly, but only two final assembly fixtures. This gives a balanced output since the operators at the two final fixtures can cope with the production from the other fixtures.

There are many interesting features in the layout and equipment for the roof and rear quarters assembly section. To begin with there is the clever design of the various fixtures. As on the front assembly fixtures, great use is made of toggle clamps to allow the work to be quickly secured in position. Hydraulic clamping is also used in certain positions. In every case, the design is such as to give maximum convenience for the welding operator. Great care has also been taken to ensure that the welding shall cause a minimum of distortion so that the work can be removed from the fixture without difficulty. It is almost needless to say that a high degree of accuracy in the work as delivered from the press department and also in the sub-assemblies is absolutely essential to the efficient use of these, and indeed of all the other fixtures in the body building section.

Some of the welding equipment that has been specially designed for use on the roof and rear quarters assembly calls for brief comment. Probably the most interesting item of equipment is a portable seam welder used in welding the roof channel to the roof panel side. To the best of our knowledge, Vauxhall Motors Ltd. is the only organization to develop a fully successful portable seam welder. The advantages of such a tool are too obvious to need stressing.

Considerable use is also made in this section of welding guns that are mounted in a ring frame in such a manner that the electrode holders can be swivelled through 360 deg. This type of gun, of which an illustration is given, is particularly useful for welding round window apertures since it allows the best and most convenient position to be maintained for every spot weld. Mention may also be made of the heavy shields that are incorporated in the fixtures at the points where gas welding is to be carried out. One of these shields can be seen in the illustration of the roof panel to rear quarters fixture. The purpose of the shield is to retain the heat in the welding area. There are two reasons for this: first, it helps to ensure a sound weld; second, it minimizes the area affected by the heat of the torch. The shield shown in the illustration is hydraulically swivelled on to and away from the work. In the one position it forms a very effective clamp.

The final assembly of the body shell is carried out on a closed circuit conveyor. Front and rear underbody assemblies are delivered by conveyor from the press weld section. On reaching the body building section they are mounted on a bogie fixture on a closed circuit conveyor. These assemblies are welded together and certain minor items are gun welded on while the work passes along the first run of the conveyor towards the main body welding fixture. At the appropriate point the bogie, and with it the underbody, leaves the conveyor and the front assembly and the roof and rear quarters assembly are mounted on the underbody. The part assembled shell is then pushed into one of three main body building fixtures. These, as with other welding fixtures, have been designed to allow easy loading, accurate and secure location, easy access to the welding positions and rapid withdrawal of the welded shell from the fixture.

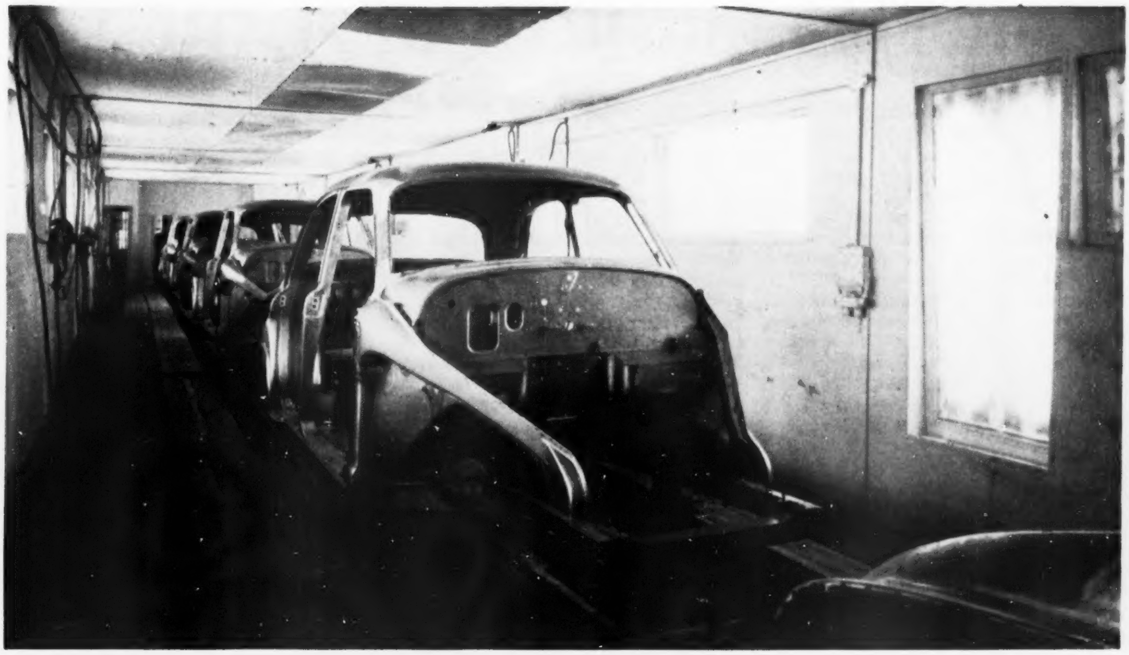
When the shell is welded, the bogie and with it the shell, is pushed out from the other end of the fixture back on to the conveyor. During the circuit this conveyor runs at varying heights so that it is always at the most convenient height for the operation that is to be carried out. All the necessary gun or gas welding equipment is mounted at appropriate points round the conveyor circuit. At the end of the circuit the body shell is lifted from the bogie by an electric hoist and transferred to another bogie on the metal finishing and door hanging conveyor line. The shell bogie is then reloaded and starts on the circuit again.

The operations carried out on the body shell during its passage down the metal finishing line do not call for detailed comment. It may, however, be pointed out that the solder paint method of tinning is employed for filling all gas weld joints. This method gives remarkably good adhesion and has the added advantage that the solder has only a low tin content and is therefore economical. Furthermore, the surface produced is one that requires very little filing. Perhaps the outstanding feature of this line is the relatively small amount of metal finishing that is necessary. In the main this may be ascribed to three factors :—

(1) The production of press work that is free from surface blemishes.
(2) The care taken during both inter- and intra-department handling.
(3) The care taken to avoid damage during the sub-assembly and assembly operations.

There is not a great deal of welding carried out on the metal finishing and door hanging line, but in at least two cases spot welds have to be made in

The main fixture for the body shell



A sanding booth in the metal finishing line

positions that with ordinary gun welding would leave an exposed welding mark that would have to be removed. To avoid damaging the exposed surface, a poker gun is used instead of the normal pincer gun. The circuit for poker welding is completed by a copper strip remote from the exposed surface. At the end of the metal finishing line, the body shell, complete with doors and luggage trunk lid, is lifted from the bogie and transferred to an overhead conveyor for transport to the painting department. The bogie then continues along the conveyor to the start of the metal finishing line.

The methods and equipment used for producing Velox and Wyvern bodies are outstanding examples of the great improvements that have been made in the British automobile industry during recent years. In connection with press welding, Vauxhall Motors Ltd. had the advantage of being able to draw upon the experience of the General Motors Corporation, but the actual development of the machines was the responsibility of the Vauxhall engineers, who had to consider problems that are not encountered in America. For example, the need to employ batch production in the section for doors and luggage trunk lids, meant that considerable thought had to be given to the question of loading the welding units into the presses. This problem was successfully solved and the change-over can be made during the normal day-shift without the press weld section falling behind the press section.

Of other production equipment the outstanding items are the welding guns and the welding fixtures which are completely designed and manufactured by Vauxhall Motors Ltd. As was stated earlier, every gun is specifically designed for a specific purpose.

The lay-out of each section is remarkably efficient, particularly when it is remembered that the various sections described in these notes had to be installed in existing factory buildings. Without any doubt the most spectacular development is the integration of the press section and the press-weld section for the production of doors and luggage trunk lids, but this layout should not be allowed to overshadow the neat and compact arrangements for front assemblies, roof and rear quarters assemblies and final assembly in the body building section.

Finally, mention must be made of the work of the materials handling division. Pressings and sub-assemblies in a wide variety of shapes and sizes had to be transported over relatively long distances, and many of the parts were susceptible to damage. The system in use was adopted only after a comprehensive analysis of all the factors, including the effects on final cost, which after all, must take precedence of all other factors except work quality.

TRADER HANDBOOK

THE 1953 edition of the *Trader Handbook* provides nearly 500 pages of essential information for manufacturers, trade suppliers and repairers of motor, motor cycle and cycle goods. All the contents have been checked and brought up-to-date. The practical and comprehensive treatment of every aspect of these industries provides useful information for the daily problems of buying, selling and servicing. There is no doubt that this handbook will also prove of very great value to overseas firms seeking contact with British suppliers.

A new feature of interest to the motor vehicle service engineer is a table of " front end " service data that gives castor angles, camber angles, king pin inclinations and toe-in for various makes of vehicles.

Specification details in the technical section include data on private cars, goods vehicles, buses and coaches, battery electric vehicles, agricultural tractors, three-wheelers, motor cycles and auxiliary engines for pedal cycles. In this section the specifications for more than 800 vehicles are analysed and set out in conventional form. In the Buyer's Guide section the space given to suppliers of motor vehicle spare parts has been greatly enlarged to include a great amount of additional information.

This indispensable Handbook is published by the Trader Publishing Co., Ltd., Dorset House, Stamford St., London, S.E.1. It is priced at 12s. 6d.

METALLURGICAL RESEARCH

Some of the Work of the British Non-Ferrous Metals Research Association

ALTHOUGH the greater part of the metal used in motor vehicle construction is steel, there is nevertheless an appreciable amount of non-ferrous metals in each vehicle. Therefore, research leading to improvement in this class of metals is of concern to the industry. Reliance is generally placed on material suppliers to provide metals to suit the requirements of the motor industry. However, it is still the responsibility of the motor manufacturers to ensure that these materials come up to the required standard in every respect. Part of the work of The British Non-Ferrous Metals Research Association is concerned with the development of better methods of material control.

Chromium plating

The quality of chromium plating used on motor cars produced both in this country and abroad has been the subject of much criticism in the post-war period. The reason why the plating on products manufactured during this period is not so durable as that produced earlier may well be that the processes used for the production of bright nickel deposits produce results which are less resistant to service conditions than those obtained from the older types of plating bath. There are a number of processes for bright nickel plating, most of them of a proprietary nature. These processes are all relatively new and experience is showing that the deposits produced from them may fail prematurely in service, although they conform to the existing specifications and will pass the corresponding tests. One of the aims of the Laboratories is to investigate these problems so that better inspection procedures can be formulated for quality control of chromium plated parts.

The disadvantage of the older methods is that they give a matt finish to the nickel plating. In production it is therefore necessary to remove the components from the plating line for polishing, and then return them to the line for their final chromium plate. This additional operation is not necessary with bright nickel plating. The wide application of the bright plating process is mainly a post-war development. Bright deposits are obtained by adding to the nickel sulphate bath certain proprietary organic substances in relatively small quantities. The effect of these small additions is to alter the structure of the deposit in ways which are not yet fully understood. Consequently it is not at present known why these deposits are not so satisfactory as ordinary matt ones, but it is possible that a certain amount of porosity is introduced by the inclusion in the coating of some of the additives.

When nickel was in very short supply, a considerable amount of work was done to find substitutes. Over a thousand samples were exposed in a number of different atmospheres, including those to be found in coastal and industrial areas. An entirely satisfactory substitute for nickel has not yet been found, but the tests are still being continued in case the shortage should again become more acute. One of the results of this work was that defects were discovered in certain types of nickel plating and useful new information emerged concerning the influence of deposit thickness. Copper may be substituted for a certain proportion of the nickel plate.

It is well known that the quality of the finished product depends largely on the thickness of the nickel plating and that chromium is applied only to prevent the tarnishing of the nickel. Thick deposits of chromium would protect the underlying metal if they were continuous, and the nickel could be dispensed with, but they cannot be used in this way because they crack or plate dull. An investigation is in hand on this subject. In nickel plating, faulty operation of the plating bath can lead to particles of various kinds becoming suspended in the electrolyte and being deposited on the plate, to give a rough finish. When tanks lined with certain mixes of rubber are employed, the electrolyte may also be contaminated by some of the constituents of the rubber.

Probably the only sound method of investigating the cause of defects in bright nickel deposits is to set up a pilot production line in the Laboratory and operate it at capacity for some time. By using such a pilot plant, in which strict control can be maintained over all the variable factors, much more reliable estimates of the causes of defects may be made than is possible either in full scale production plants, which are not equipped for accurate control of variables, or in the Laboratory under conditions which must in many respects differ widely from those obtained in practice.

Apart from visual inspection, the tests currently in use for checking the quality of nickel/chrome plate fall into two categories. Both involve the destruction of the plating or the component and can therefore be applied only to a small fraction of the total output of components. In the first of these categories are tests for thickness of deposit, and in the second, accelerated performance tests under specified corrosive conditions. The most widely used thickness test is the well-known B.N.F. Jet Test in which a jet of appropriate corrosive solution is directed against the plated surface and the time taken to penetrate the coating in question is noted. The thickness is then read from a calibration chart. The

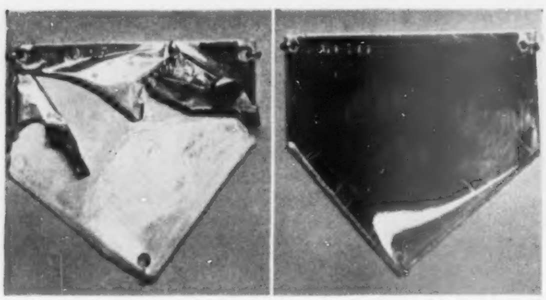


Fig. 1. In the electrolytic process for plating aluminium, adhesion before stoving is poor, as may be seen on the left, but it is satisfactory when the component has been stoved as on the right

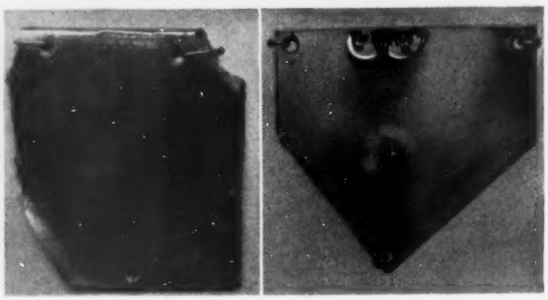


Fig. 2. Samples of aluminium plated by the process developed by the B.N.F. Laboratory. Left, D.T.D.424 welded to commercial grade aluminium; right, aluminium with a copper rivet in the centre

standard method of accelerated performance test is to spray the article, either intermittently or continuously, with salt solution for a period of 96 hours. The results obtained are, however, very difficult to interpret reliably, and efforts are being made, based on the experience gained from the atmospheric exposures, to find a better acceptance test.

In applying the jet test to dull nickel plating, a linear relationship between the time of penetration and the plating thickness holds good, but with bright nickel the linear relationship does not apply. However, investigations have been carried out recently on this subject and a suitable conversion factor, which varies with the thickness of the plating, has been obtained and reliable results can now be obtained when using the jet method for testing bright nickel plating.

Chromium plating on aluminium

Hitherto two methods have been used for chromium plating on aluminium. In both methods the first step is to degrease the component. Then the operations are as folllows: In the zincate immersion process, the component is pickled in a nitric hydrofluoric acid bath. Zinc plating is then applied by the immersion process, and is followed by copper plating and then by nickel plating. Between each process the component is, of course, rinsed. Finally it is polished and then chromium plated.

In the electrolytic process, the component is first given a cathodic clean, and then an acid dip followed by another cathodic clean. It is next given a zinc plate by the electrolytic flash method, followed by a brass plate and then a nickel plate. Again, a rinse is given between each of these stages. Finally, it is stoved and polished, and then chromium plated. Two samples of nickel plated aluminium sheet are shown in Fig. 1. The lower corners have been broken off each sample to test the plating. The first sample was tested in this way before stoving, and the plating was easily peeled off. In the second sample, which was tested after stoving, scratch marks may be seen at the broken edges where unsuccessful attempts have been made to lift off the plating.

Both plating methods have their disadvantages. With regard to the zinc plating operations, the immersion process gives a thick film of zinc which is thought, in some quarters, to be spongy and to result in poor resistance against corrosion and subsequent failure of the plating. Moreover, it is impossible to plate satisfactorily components made up from different alloys of aluminium, or held together with copper rivets and bolts. On the other hand, the electrolytic flash method gives a thin, incomplete zinc coating which again cannot be regarded as being entirely adequate.

The Laboratories have developed on a pilot scale a new method which combines the good features of the other two, and which may be used in conjunction with bright nickel plating to eliminate the polishing operation. This process is as follows: After being degreased and given a cathodic clean, the component is acid dipped and again cathodically cleaned. Next it is given an immersion zinc plating, which is then stripped off in a nitric acid bath. This activates the aluminium surface so that a more uniform deposit may be obtained in the next stage, which is zinc plating by the electrolytic flash method.

After this, the component is brass plated and one of the two following procedures may be adopted. Either it is nickel plated, polished and chromium plated, or it is bright nickel plated and chromium plated. As before, each stage in the plating process is followed by a rinse. In Fig. 2 may be seen two test pieces, one made up of a strip of D.T.D. 424 welded to commercial grade aluminium, and the other is a piece of aluminium with a copper rivet in the centre. In both these samples the plating adhesion is strong.

Fig. 3. The etched surface of aluminium magnified approximately 12,000 times horizontally and 1,200 times vertically

The electron microscope

One of the instruments used in research on plating as well as in other work is the electron microscope. There is a limit beyond which the lens of an ordinary microscope cannot resolve the structure of any object. However, if a beam of electrons is used instead of light, the resolving power is much greater because the wavelength of an electronic beam is about 100,000 times shorter than that of light. Unfortunately, because smaller lens apertures must be used, the resolving power is only about 100 times that obtained with light. However, this is still a marked improvement. Electron beams have a similar effect as light on a photographic plate, and they may be used to illuminate a fluorescent screen.

There are two methods of using an electron beam to photograph plating or other surface characteristics. One is to apply to the surface a weak solution of collodion, or any other suitable

Fig. 4. Zinc plating after 10 sec of the electrolytic flash process. Coverage is not very good

Fig. 5. Although the plating obtained by the Zincate immersion process is thick, it is not even

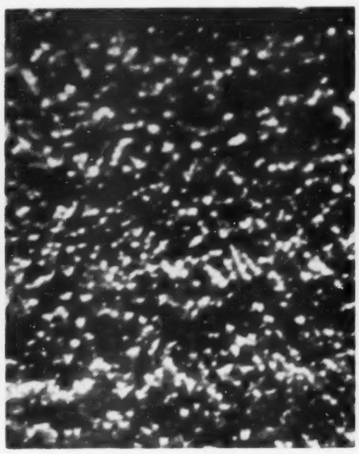


Fig. 6. An even and continuous deposit is obtained with the new process developed by the B.N.F. Laboratories

plastic, and then when it is dry, to remove the thin film of plastic and shine the electron beam directly through it. The electrons are scattered in proportion to the thickness of the film at each point so that a picture of the surface will be reproduced on a photographic plate positioned on the side of the film opposite to the electron beam projector.

Since the film will be thinnest where high points occur on the surface under examination, a greater proportion of the electrons will pass through at those points. Thus, on the photographic positive they will appear lighter than the low points. The contrasts outline the structure of the surface from which the plastic copy was taken, and it therefore appears much as it would have done had it been examined with a light microscope, but the details are much clearer.

The second technique adopted in using the electron microscope is one involving photographing the actual surface illuminated by an electron beam instead of light. Because of the low penetrative power of electrons and the fact that they are absorbed by what they strike, there would be little or no reflection of a beam directed normally on to the surface, and under these conditions photography is impossible. However, if the beam is directed at an acute angle to the surface, reflection does occur, and the reflected beam may be used for photographic purposes. In this case, electrons striking peaks on the surface are either absorbed or scattered, so that the peaks are illuminated on one side, and a long shadow is thrown on the other side. This gives an effect similar to that obtained, on a much larger scale, by the sun setting over mountainous country.

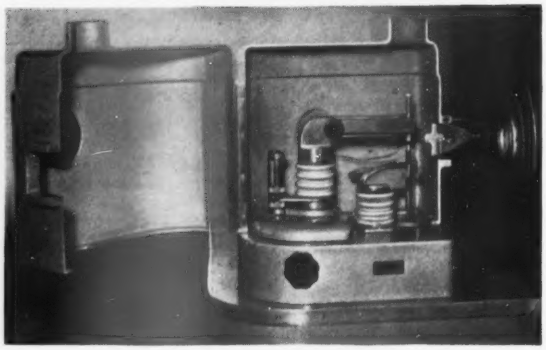


Fig. 8. The spectrographic spark stand of the Production Control Quantometer

Illustrated in Figs. 3, 4, 5 and 6 are some photographs takes with the electron microscope by this reflection method. The photographs show the differences between the three methods of zinc plating which may be used, as already described, in preparation for chromium plating on aluminium. Fig. 3 shows the etched surface before plating, and Fig. 4 shows the zinc particles deposited on the surface after 10 seconds of the electrolytic flash method of plating; the appearance is not much changed when the process is complete after 10 seconds. Fig. 5 illustrates the completed plating obtained after three minutes of the zincate immersion process, and Fig. 6 shows the deposit obtained by the new process developed by the Laboratories. It is easy to see that the new method gives a much more even and continuous zinc deposit than the other two.

The Production Control Quantometer

The Production Control Quantometer, Fig. 7, is a fairly new instrument which has been developed and is at present being manufactured by the Applied Research Laboratories, Glendale, California. It is a high speed spectrograph and the one in use at The British Non-Ferrous Metals Research Association's Laboratory is set up primarily for the analysis of copper and its alloys. However, it can also be supplied set up for carrying out analyses of steel or aluminium alloys. Certain Continental and United States motor manufacturers are making use of this type of instrument for quality control of materials.

The principle of the Quantometer is as follows: Electrodes made from the material to be analyzed are set up in a conventional spectrographic spark stand, Fig. 8. Light from the electric discharge falls on a concave diffraction grating ruled with 24,400 lines/in. The diffracted beams of the spectrum so produced are focused on slits arranged round the arc of a circle in such a way as to select the wavelengths on which the spectrographic analysis of the material is to be based. The light that passes through each slit falls first on a concave mirror, in the form of a sector of a cylinder, which focuses it on to the cathode of a multiplier photocell. The primary photo-current so produced is amplified up to about a million times within the cell, and the amplified current passes from the spectrograph through a cable to one of a series of electrical condensers in the recording console of the equipment.

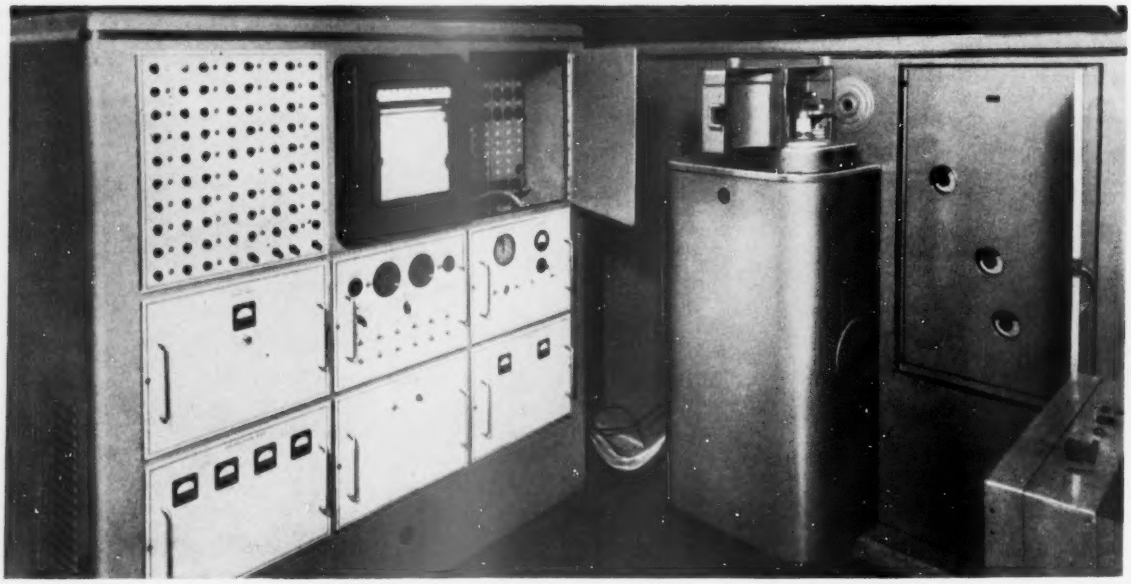


Fig. 7. The Production Control Quantometer is simple to operate and it may be used to make a complete analysis in only 2½ min of about 15 elements in a sample

As exposure, proceeds, the voltage across the plates of each condenser rises at a rate proportional to the intensity of the corresponding spectrum line. Exposure is terminated when a predetermined voltage is reached on one condenser charged by the photo-current generated by a spectrum line of the basis metal, for example, aluminium when aluminium alloys are being analyzed. At the end of exposure, the voltage on each of the remaining condensers is measured by a sensitive valve voltmeter of high input impedance, the voltages on the various condensers being measured in turn by an automatic stepping switch. Each voltage is recorded on a chart, and from the recordings the percentage of each constituent of interest may be directly determined. These percentages are ascertained by comparison with a previously prepared set of calibration curves derived from results obtained with chemically analyzed standard samples tested in the Quantometer.

With the aid of this equipment it is possible to make, within the space of 2½ minutes, a complete analysis for up to, say, 15 constituent elements, including alloying additions and impurities. The saving in time as compared with that necessary for orthodox chemical and spectrographic methods employing photography is obviously considerable. Thus, a more complete check may be made of incoming materials, or analyses may be carried out, for instance, before pouring the contents of a furnace. The accuracy of the analysis is generally higher than that obtained by more conventional spectrographic methods.

Fig. 9. A jet test machine

Other work

Other work that has been done or which is in progress includes investigations into stretcher-strain markings and the surface condition known as orange peeling, which occur on light alloy sheet, particularly aluminium-magnesium alloys. Enough is now known about these phenomena for recommendations to be made with regard to material specifications for their avoidance. The orange peel effect is due to large grain structure in the metal. On the other hand, stretcher-strain markings are characterized by the formation, on pressings, of distinct markings, often visible even after painting. They are due to effects associated with deformation taking place in the grains. The results of this work are to be published for the first time later this year in a paper to be read before the Institute of Metals.

Investigations have also been carried out into the prospects of wider use of electropolishing processes. The laws governing these processes have been determined, and it has been concluded that whatever process and solution is used, electropolishing alone will only brighten, but not give a perfectly smooth surface. It is very suitable for polishing intricate shapes that cannot be treated by mechanical methods. A special technique has been developed by the Laboratory whereby it is possible to arrive quickly at a cathode arrangement which will avoid the excessive rounding off of any sharp corners whose dimensions must be preserved accurately.

Much work has been done in the general field of corrosion of non-ferrous metals in various environments. X-ray and electron diffraction equipment is available in the Laboratory for the analysis of films on corroded metal surfaces as well as for other investigations. The electron diffraction apparatus was designed and

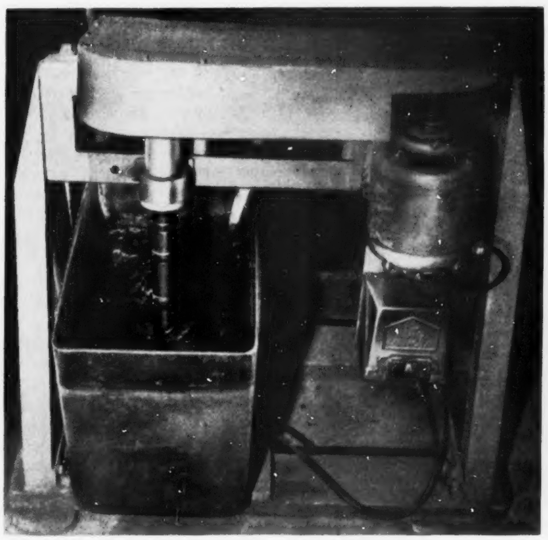


Fig. 10. When the jet test is not suitable, this rig may be used for assessing corrosion resistance

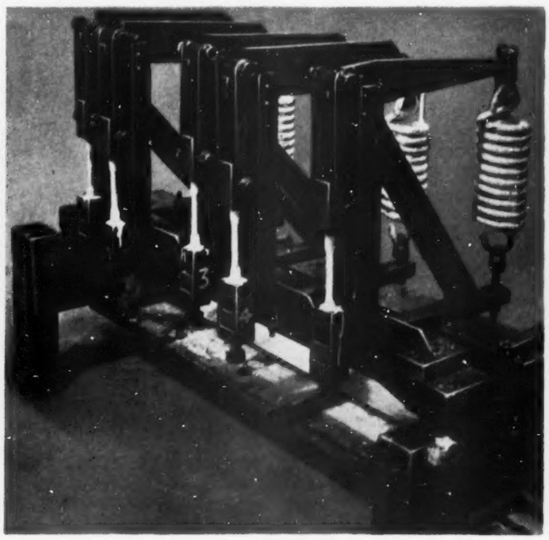


Fig. 11. A battery of machines used for tests under conditions of stress and corrosion

constructed by the Laboratory staff.

The Association has recently published the results of an investigation of the factors which may sometimes lead to the corrosion of aluminium when immersed in tap water. An examination is also being made of ways of eliminating the corrosion risk in closed systems such as radiators. Similar work is in hand concerned with corrosion in soil and in contact with road-wash.

Among the simpler test machines used in the Laboratory are the two illustrated in Figs. 9 and 10. Fig. 9 shows an apparatus designed for testing the corrosion resistance of components subject to impact from a continuous jet of sea water containing air bubbles. The electric motor driving the pump may be seen mounted on the bench in the foreground. Sea water is circulated from the earthenware tank to a central chamber above it, and then through a number of rubber tubes to jets directed at test pieces. In the illustration it can be seen that one of the jet elements has been lifted and placed on the edge of the tank. Each element consists of a bridge piece between the two legs of which is the jet. The test piece is a metal strip secured across the ends of the legs by metal clips. During the test, the element is submerged below the brine in the tank.

Illustrated in Fig. 10 is a simple test rig for assessing corrosion resistance. A vertical shaft driven by an electric motor is positioned so that its lower end is submerged in a tank of the corrosive liquid. On the submerged end are two discs between which are clamped the test pieces. When the discs are rotated the liquid flows over the test pieces. Among the liquids most commonly used are brine, sea water, rain water or normal supply water, etc. It has been found that if resistance to sea water corrosion is to be determined, substitute solutions are not satisfactory, and the sea water must be transported to the Laboratories.

Investigations have also been carried out into the behaviour of materials under the combined influences of corrosion and stress. Fig. 11 illustrates the apparatus used for this work. The test specimens, which in this case are of aluminium alloy, are loaded by the springs attached to the ends of loading beams. Unstressed specimens are suspended near the machine so that they are subject to the same corrosive influences, and the work involves the comparison of the effects on both loaded and unloaded specimens. The humidity of the room in which the apparatus is housed is kept at about 80 per cent by a fan circulating the air in the room over a water tank. Twice a day the specimens are sprayed with brine, and the humidity is enough to ensure that they remain damp over the whole of the time between spray applications.

The Liaison Department

One of the outstanding features of the organization of the Laboratories is the Liaison Department. It is staffed by several graduates under the supervision of the Chief Liaison Officer. One of its main functions is to ensure that executives of member firms are kept aware of the facilities that are available at the Laboratories for investigating and solving their difficulties. It also ensures that the research staffs are kept well informed with regard to the practical problems and new developments that arise in industry, so that there is close collaboration between the research staff and those concerned with the practical application of the research projects.

Its principal duty, however, is to provide a free consulting service to members. Enquiries received fall into three general categories. The first is enquiries relating to design, for example, questions are asked as to what materials are most suitable for certain applications, or concerning the properties of different alloys. The second category concerns failures. Often manufacturers feel it advisable to call for an independent opinion as to the cause of failures that they have experienced; the Laboratory staff, by virtue of their intimate knowledge of metallurgical problems, and also because of the extensive sources of information readily available to them, are well qualified to give such an opinion. Production enquiries constitute the third category. These cover a wide range of production problems, including electroplating, foundry work, machining, stamping, pressing, etc.

The Information Department and Library works in close co-operation with the Liaison Department. A very complete collection of all published information on non-ferrous metallurgy and related subjects, including patent literature, as well as much unpublished material, is in the library and available to members. In 1952, over 9,500 loans were made from the library. In addition, translations are provided of the more important foreign papers of German, French and Russian origin. Literature searches on required topics are also made.

A Bulletin is sent out each month to the members. It contains a large number of classified abstracts and notes from the world's technical press. Also distributed to members is the B.N.F. Review, which contains information concerning current researches of the Association. The Review is presented in such a manner that it may be of the greatest possible value in directing the attention of executives to the practical application of the results of the investigations. In other words, it gives results and reasons, instead of dealing at length with the whole experimental procedure. The full reports, which describe the researches in detail, are available to members on request.

ALUMINIUM BONDING

ORIGINALLY developed for heat transfer applications, aluminium bonding consists in bonding molecularly pure aluminium or any of its common alloys to cast iron, Inconel, Nimonic or titanium. The three basic steps in the process are: surface preparation and masking of the base metal; dipping the part into an aluminium bath to produce a bonded coating; casting the aluminium alloy against the prepared surface. In applications requiring heat-treated steel members of high strength, it is advisable to use the more metallurgically stable steels, for example SAE 4130 or 4340, which will not be affected by the temperatures encountered in the Al-Fin processing.

Ferrous metals, nickel and titanium react with molten aluminium to form an intermetallic compound which provides the bond between the aluminium alloy and the base metal. The bond has a mean tensile strength of 15,000 lb/in^2 and withstands shearing stresses of 8,000 lb/in^2. It also acts as a chemical buffer between the two metals, thus minimizing the danger of electrolytic corrosion. There is no measurable resistance to heat flow across the interface of the two bonded metals.

In addition to air-cooled cylinder barrels and commercial heat exchangers, the field of heat transfer applications of the Al-Fin process has been extended to internal cooling fins of eddy-current clutch rotors and exhaust pipe heat exchangers in aircraft engines. Important weight reductions in aircraft and automotive construction have been effected without sacrificing strength, for example, in aluminium housings with bonded-in steel bearing retainers, aluminium timing gears with bonded-in steel hubs and in pistons with bonded-in ferrous ring carriers. Steel-backed aluminium bearings are suitable for heavy duty service. The process can also lead to savings in production and assembly costs.

For high temperature service, an allowance must be made for thermal stresses arising from uneven expansion of the two elements. Thermal stresses can be compensated by: (a) designing the unit so that one member has thickness/length ratios which permit elastic deformation, (b) using ferrous metals, such as Ni-Resist type I cast iron, whose coefficient of thermal expansion is similar to that of aluminium, or (c) using a low-strength, high-ductility aluminium alloy which will offer little restraint. (*M.I.R.A. Abstract* 6189.)

MANUFACTURING TOLERANCES

The Pilot + 2 Limit System

By F. W. M. Lee, M.I.P.E.*

A NEED for a comprehensive limit system covering the whole field of engineering has long been apparent. It was felt, however, that even if such a system could be evolved, it would be both unwieldly and complicated. In a short article it is not possible to make reference to the research that was undertaken in arriving at the final design of the Pilot + 2 System, except to say that many systems were analyzed and superimposed upon one another graphically to see if they had any common features. There were none and it became abundantly clear that any proposed system must be based upon a new conception in keeping with modern requirements.

The first essential was to determine the practical magnitudes of engineering tolerances in use to-day. For this purpose nearly 10,000 limits on gauges of recent manufacture produced by The Pilot Plug Gauge Co. Ltd. and John Harris (Tools) Ltd., Warwick, were analyzed. As a result of this investigation the original conception of the Pilot + 3 System was modified to Pilot + 2. A wealth of information was obtained concerning the limits used in very many branches of engineering, and with these tolerances in mind, a system was constructed upon a number of practical ideals. In what follows, the ideals are not necessarily placed in order of importance, but are so arranged that they will assist an understanding of the logical development of the system.

Ideal No. 1

Should the system be unilateral or bilateral?

There should be little doubt as to the correct choice in this connection for the following reasons. The unilateral system is based upon a standard hole and the diameter of the shaft is varied to give the requisite fit, as opposed to the bilateral system in which the shaft is made to a standard and the hole varied to provide the fit. For engineering reasons it is easier to vary the diameter of a shaft than it is to alter the size of a hole, and this very practical fact decides it must be unilateral. In support of this conclusion the dominating tolerance systems are unilateral.

The Pilot + 2 System is therefore unilateral, by which is meant the smallest diameter of the hole is never less than the basic diameter, and the tolerance or the amount of error which will be tolerated, is always in a plus direction, that is, the largest diameter to which the hole may be made is greater than the basic size of the hole by the tolerance.

Ideal No. 2

The tolerances provided must be suitable for the whole field of engineering practice

As already mentioned, nearly 10,000 gauge limits were analyzed and details of this task cannot possibly be shown here, only inspection of the chart constructed in accordance with Ideal No. 3 will indicate whether tolerances have been provided for any particular product. It is felt that this ideal has been amply catered for.

Ideal No. 3

It must be simple

By this is meant that :—

(a) Any tolerance for any class or diameter of hole can be calculated mentally.

(b) The table for the smaller range, of which 98·4% of limited holes are below 4 in in diameter, can be memorized.

(c) The fundamental basis of the system can be explained to an apprentice in a few minutes.

The tolerance table

Table I shows the tolerance table for classes 1 to 6 and diameters from 0 to 6 inches. It is based upon simple calculations which ensure that as the diameter of the hole increases the tolerance also increases, and as the class of the hole becomes progressively coarser the tolerance steps also become coarser. The table is constructed upon three essentials :

(1) The + 2. of the Pilot + 2,

(2) The class or quality of the hole,

(3) The diameter,

and the interplay of these three factors ensures that the above-mentioned requirements are fulfilled.

Calculating the tolerance

To the inch diameter of the hole add the + 2. of the Pilot + 2. and multiply by the class number of the hole.

A 1 in diameter class 2 hole tolerance is therefore—

1 in diameter + Pilot + 2. = 3 times the class 2 = 0·0006 in tolerance; or again 1 in + 2 = 3 × 2 = 0·0006 in tolerance.

A 2 in diameter class 4 hole tolerance is—

2 in + 2 = 4×4 = 0·0016 in tolerance.

A 20 in diameter class 2 hole tolerance is—

20 in + 2 = 22×2 = 0·0044 in tolerance.

For diameters of 0·125 in and less the class number is the *tolerance in tenths*.

For holes above 0·125 in and not 1 in diameter the tolerance is obtained thus:—

0 + 2 = 2× the class number = the tolerance in tenths.

TABLE I—PILOT + 2 TOLERANCE SYSTEM FOR HOLES

	CLASS	0 but not 0·126"	0·126" but not 1"	1" but not 2"	2" but not 3"	3" but not 4"	4" but not 5"	5" but not 6" →
	↑							
	6	6	12	18	24	30	36	42
BLACK	5	5	10	15	20	25	30	35
AMBER	4	4	8	12	16	20	24	28
BLUE	3	3	6	9	12	15	18	21
GREEN	2	2	4	6	8	10	12	14
RED	1	1	2	3	4	5	6	7

UNIT—0·0001 in

Examination of the table shows that for holes :—

0 — 0·125 in the tolerance increases with class in steps of 1 tenth.

0·126 in — 0·999 in the tolerance increases with class in steps of 2 tenths.

1 in — 1·999 in the tolerance increases with class in steps of 3 tenths.

1·999 in — 2·999 in the tolerance increases with class in steps of 4 tenths and so on. The table can therefore be memorized, and the method of obtaining the tolerance explained in a matter of minutes.

It will be noted, apart from the modification of 0·125 in downwards ;

*Managing Director, The Pilot Plug Gauge Co. Ltd., Coventry.

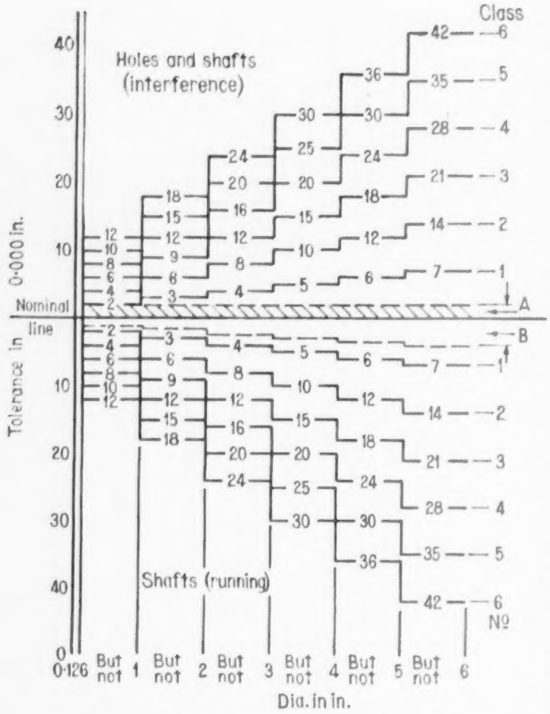


Fig. 1. Pilot + 2 limit system for holes and shaft.
A—Maximum manufacturing tolerance for "go" end
B—Maximum wear allowance for "go" end.

which is incorporated for the use of the instrument, watch, clock, and similar trades, that the value of the tolerance changes at nearly 1 in intervals (1 in to 1·999 in) fractional sizes of an inch being ignored. The soundness of this method cannot be elaborated here.

It will thus be seen that the Pilot + 2. System tolerances can be calculated mentally, can be memorized, and because of its simplicity be explained to an apprentice in a matter of minutes.

Ideal No. 4

The total tolerance range must be kept to a practical working minimum

Table I indicates that whilst only six classes of tolerances are shown, there is no theoretical limitation to the number of classes provided. Industry, however, has already decided this criteria, for it has been found by investigation that classes 3, 4 and 5 are the most popular. This is a general assertion and must be modified to suit the particular product. In one case, classes 2, 3 and 4 may be stipulated by a manufacturer as suitable for his design of machine tool, whilst in another firm with less exacting requirements the coarser tolerances of 3, 4 and 5 could be used.

The range must therefore be decided by the individual firm concerned. Here it might be suggested that if a firm, bearing in mind the article produced, lays down that only classes 3, 4, and 5 may be used, but a tighter tolerance is considered essential, it may be specified only by special permission from the Design and Production Department. In this way, limits which are unnecessarily tight and have no bearing upon design or production requirements will be avoided, and prevented from being perpetuated and increasing production costs.

Ideal No. 5

The system must provide tolerances for shafts

Fig 1 shows the complete layout of the Pilot + 2 System as applied to shafts in which it will be noted that class tolerance lines exactly similar to the hole tolerance lines above the basic line are provided below the basic line. These shaft lines are calculated in precisely the same manner in which the hole tolerances were decided, so that the tolerance table, Table I, could literally be turned upside down and serve two purposes. The question of shaft tolerances will not be pursued here, as a full explanation with examples will be given under Ideal No. 6 concerning fits.

Ideal No. 6

The system should give a mental picture of fits

Here it would be as well to recall that the Pilot + 2 System is a unilateral System, in which the smallest diameter of the hole—subject to qualification later—is never less than the basic diameter. In any fit, the smallest diameter of the hole is the dominating dimension, so it is the nominal or basic diameter of the hole which must be kept in mind when deciding a fit. To make this clear, three types of well defined fits will be considered.

Running fit

It is assumed a 1 in diameter hole has been produced to its smallest size of exactly 1 in in diameter, and a shaft is required to run in this hole with a minimum clearance of 0·0003 in, and a maximum clearance of 0·0006 in. Reference to the class lines below the basic line for 1 in diameter indicates that this particular shaft can be described by the figures 1 and 2, of which 1 is 0·0003 in below the nominal line, and 2—0·0006 in. In the Pilot + 2 System the nominal line is indicated by means of an oblique stroke /, and since the shaft of a running fit is less in diameter than the smallest or nominal size of the hole, the class numbei controlling the size of the shaft would be placed to the right, or minus side of it thus, /12.

Because the largest diameter of the hole in the unilateral Pilot + 2 System is controlled by the class number, or in other words the tolerance, the hole is then perfectly described in the case of a class 1 hole by 10/. The 0 or zero is the nominal size irrespective of class or diameter, from which it follows that only one number, the class number, is required to describe the hole, since the other number is always zero and can be ignored.

The running fit can now be written 1/12, of which the first 1 is the class 1 hole, and the 12 a shaft which is a minimum of 0·0003 in, and a maximum of 0·0006 in smaller than the nominal size, and therefore has the desired clearance.

Interference fits

In a similar manner, an interference fit would be written 123/ in which, by usage, the first number is the class 1 hole and 23 the shaft. In this case, the shaft is to the left or plus side of the nominal line, which means it is bigger than the hole because the class numbers controlling its size are numerically greater than the class number of the hole.

Transition fit, or selective assembly

By the same reasoning, a selective assembly fit is written 110/ from which it is immediately apparent that the hole and the shaft have the same tolerances and must therefore be mated. The fit of the hole described in the above manner should be incorporated in the assembly or subassembly drawing, and the actual tolerances stated on the component drawing.

Ideal No. 7

Provision must be made for ball and roller race housing tolerances

Ball and roller race housings require three types of holes:

(1) Oversize.
(2) Transition or bilateral.
(3) Undersize holes.

The Pilot + 2 System, being essentially a unilateral System, would not be complete if it could not provide suitable tolerances for ball race housings. Before proceeding it is necessary to define the above types of holes.

Oversize. The maximum permissible diameter and the minimum diameter of the holes are both larger than the nominal size of the ball race to be used in it.

Bilateral. The maximum size is greater and the minimum size less than the nominal diameter of the ball race.

Undersize. Both diameters are smaller than the nominal outside diameter of the ball race.

With an oversize hole the smallest

diameter of the hole is larger than the basic diameter, and therefore zero or O will not describe the minimum diameter. A number, which is the class number, is used to define how much the smallest diameter is larger than the basic diameter and therefore an oversize hole requires two numbers to define it. Since two numbers are used to describe a shaft in the Pilot + 2 System, a prefix "B" for bearing is used to designate a bearing hole; an example will make this clear. It is required to describe a bearing hole for a 1·5 in outside diameter ball bearing with an easy sliding fit of + 0·0024 in top limit, and + 0·0012 in bottom limit. From memory, this will be a class 8 and class 4, or by calculation, 24 divided by 3 = 8, 12 divided by 3 = 4. Because both are plus and therefore to the left of the oblique stroke, the bearing hole would be described accurately by B.8.4/. In a similar manner a bilateral hole could be described by B 2/1, and an undersize hole by B / 1·4.

One of the salient features of the Pilot + 2 System is the mental picture conveyed by the position of the class numbers relative to the oblique stroke or basic line, indicating which of the three categories the hole is in, and the magnitude of these numbers immediately shows the type of fit between the bearing and its housing.

Ideal No. 8

The manufacturing tolerances for gauges must be stated

The usual practice of allowing the Gauge Maker 10 per cent of the limit as a manufacturing tolerance in a positive direction on the "Go" end is adopted; the 10 per cent manufacturing tolerance should be split on the "No Go" end as shown in the following example. The gauge to be made is nominally 1 in in diameter with a 0·001 in, or in other words, 10 tenths limit. Since 10 per cent of the tolerance can be allowed in the manufacture of the "Go" end, and as already mentioned, this is used in a positive direction, the "Go" end can be to a minimum size of exactly 1 in in diameter, and a maximum size of 1 in plus 0·0001 in.

TABLE III—REPLACEMENT CHART, B.S.I. "U" by Pilot + 2

B.S.I. "U" DIA. FROM	B.S.I. "U" DIA. TO	PILOT +2 CLASS
0″	0·125″	6
0·126″	0·290″	3
0·300″	0·590″	4
0·600″	0·999″	5
1·000″	1·490″	4
1·500″	1·999″	5
2·000″	3·999″	4

that production has the greatest possible tolerance.

"Workshop" and "Inspection" gauges

No mention has been made of "workshop" and "inspection" gauge tolerances, for the reason that they are considered to be correct in theory, but impossible in practice. A small excursion will make this point clear. A new "Workshop" "Go" end is very slightly larger than its "inspection" counterpart, but owing to the severer conditions under which the "shop" gauge is used wear is more rapid, with the result that it is smaller than the "inspection" gauge in a very short time, thus completely reversing this consideration. It may be argued theoretically that frequent inspection of the gauges will prevent this, but in practice it must be admitted that such an ideal state is seldom, if ever, attained. With this in mind and the fact that sharing the tolerance between "inspection" and "shop" robs the operator of much needed assistance the Pilot + 2 System employs only one manufacturing tolerance. In the chart Fig 1

TABLE II—PILOT + 2 AND B.S.I. "U" EQUIVALENTS CHART

TOLERANCE CHANGES IN RELATION TO DIAMETER.

Change Steps Pilot + 2	Change Steps B.S.I. "U"	B.S.I. "U" Tolerance	Pilot + 2 Tolerance	Pilot + 2 Class	Tolerance Difference
0		6	6	6	0
0·125″			6	3	0
	·290″	8	8	4	0
	·590″	10	10	5	0
0·999″	·990″	12	12	4	0
	1·490″	14	15	5	+1
1·999″	2·090″		16	4	+2
	2·790″	16	16	4	0
2·999″		18	16	4	—2
	3·590″		20	4	+2
3·999″		20	20	4	0
	4·490″		18	3	—2
4·999″		22	24	4	+2

0
Class 6
·125
·126
Class 3
·290
·300
Class 4
·590
·600
Class 5
·990
1·000
Class 4
1·490
1·500
Class 5
1·999
2·000
Class 4
3·999
4·000
Class 3
4·490
4·500
Class 4
4·999

B.S.I. "U" Tolerance Equivalents in Pilot + 2 for Various Diameters.

UNIT—0·0001 in

Here it must be stated that it is always the endeavour of the gauge manufacturer to make the gauge to the 1 in plus one tenth, or top limit diameter, with the object of giving the maximum wear allowance or life. Turning to the "No Go" end, the manufacturing tolerance is split, that is to say, the "No Go" end may be 5 per cent of the limit larger than its theoretical size of 1 in plus 0/001 in, or smaller than this size by 5 per cent of the limit. Here again, the gauge manufacturer endeavours to keep the gauge up to the top limit, or its greatest permissible size, in this way ensuring

of the complete Pilot + 2 System it will also be noted that the maximum manufacturing tolerance has been confined to two tenths ; this will be referred to later.

Ideal No. 9

The amount the " Go " end may be allowed to wear below the nominal size before withdrawal from service must be defined

Established practice is that the " Go " end, the wearable member, should be allowed to wear a maximum of 10 per cent below the basic size before withdrawal from service. To illustrate conditions in use, an example with comments may serve a useful purpose. A 1 in class 1 gauge has a limit of 0·0003 in, and 10 per cent of 3 tenths is 3 hundredths of a " thou ", or 30 millionths. Under these circumstances the " Go " end would be allowed to wear 30 millionths under size, or below the basic diameter, before withdrawal from use. Since a certain amount of clearance under production conditions between the gauge and the hole is required, it might justly be claimed that the hole will be, to all intents and purposes, " size ". In the case of a 1 in diameter gauge with a " thou " or 10 tenths limit the " Go " end would be allowed to wear one tenth below the basic diameter of exactly 1 in. Space will not allow further explanations, suffice to say that this method is widely adopted in the Engineering Industry.

Ideal No. 10

The tolerance system must be such that the B.S.I. " U " limits are catered for

If this ideal can be fulfilled, two important requirements are met.

(1) Interchangeability is unaffected.
(2) B.S.I. " U " gauges are not scrapped.

Table II is an equivalents chart and Table III a replacement chart of the Pilot + 2 and B.S.I. " U " limits, from which it will be noted that in the majority of cases exact equivalents exist, and where there is a difference it is small compared with the total tolerance, and of no practical consequence. Thus, by replacement, and therefore no cost for the introduction of the Pilot + 2 System the B.S.I. " U " limits can be abandoned and a tolerance system introduced giving much needed additional tolerances in a short period of time.

Ideal No. 11

The system should reduce gauge stocks

Reference has already been made to an upper manufacturing limit of two tenths on the " Go " end, and to conserve space the user's point of view only will be considered. A gauge with a 5 " thou " limit under the 10 per cent rule could have a maximum of 5 tenths, or 1/2 a " thou " wear allowance ; or in other words, it could be this amount larger than the nominal size. Again, under the 10 per cent rule, applying to the amount the gauge can be allowed to wear below the basic size another half " thou " is available for increasing the life of the gauge, or a total of 1 " thou ", which by all standards of reasonableness is too great. It will therefore be seen that if the maximum wear allowance above the basic size is confined to 2 tenths, then such a " Go " gauge end will be suitable for any class of hole whose limit is 2 " thou ", 3 " thou ", 4 " thou ", or larger.

This limitation reduces the cost of the gauging system, for it will be apparent that in the ideal situation when the " Go " and " No Go " gauges are separate as in larger sizes that one " Go " end can be used for any hole with a tolerance of 2 " thou " and above, and a series of " No Go " gauges only will be required to control the maximum size of the hole.

Ideal No. 12

For the immediate identification of classes a colour code would be an advantage

Reference to Table I indicates that red is used for class 1 gauges, green class 2, blue class 3, amber class 4, black class 5 and above. This is carried into effect by using coloured anodized aluminium handles which ensure that the gauges are recognized instantly in use and in storage.

Ideal No. 13

The system should reduce scrap by the prevention of reamers cutting oversize holes

It is well known that a new keen reamer cuts big, and let it be assumed that the hole is 1/2 in in diameter which has been cut 8 tenths oversize, or in other words, a class 4 hole. Under the colour code class 4 is amber, it follows that by painting the reamer Amber between the top of the flutes and the shank, or slipping on a coloured synthetic rubber band, the reamer is immediately labelled as one cutting an 8 tenths oversize hole. As the reamer wears and the hole becomes progressively smaller, the colour is changed at the various tolerance grades until red finally appears making it the most valuable reamer in the shop. In this way reamer life will be increased and scrap avoided.

Summary

The design of the Pilot + 2 System was only finalized after an extensive and illuminating analysis of tolerances used by a very wide cross section of Industry to-day, as shown by the records of The Pilot Plug Gauge Co. Ltd. and Messrs. John Harris (Tools) Ltd., of Warwick.

Based as it is upon practical considerations the following is a brief pointer of its advantages.

(1) Simple to understand.
(2) Tolerances easily memorized or calculated.
(3) It is practical, as only practical tolerances are used.
(4) It is acceptable to industry, as it has been constructed upon industrial requirements.
(5) It provides a range of tolerances suitable for watches to heavy machinery.
(6) The Pilot + 2 gives an immediate mental picture of fits.
(7) Special applications, such as ball race housings are amply and clearly catered for.
(8) It is a complete system.
(9) All particulars required for the control of a tolerance system in use are given in clear and concise terms.
(10) The Pilot + 2 System will cause no confusion in firms using the B.S.I. " U " limits, and the change can be made without cost.
(11) Possesses incidental advantages found in no other system.
(12) Half a minute only is required to explain the simple arithmetical fundamental basis of the Pilot + 2 System to any apprentice.
(13) The Pilot + 2 System can be completely and concisely described by one wall chart.

The Pilot + 2 System is designed to settle in a practical manner a problem which has been with us far too long.

SPLIT-CHASSIS CONSTRUCTION

MATTHEW BROTHERS of Wallington, Surrey, manufacturers of Matbro fork lift trucks and bulk loaders, have recently carried out timed tests on the stripping down, for maintenance purposes, of one of their two-ton fork trucks, which are of split-chassis construction. The chassis can be quickly unbolted in the middle, so that the vehicle is then in two parts. For operations such as clutch maintenance and adjustment, the advantages of this design are considerable since it obviates complete stripping down.

It is estimated from the timed tests that on certain maintenance operations, anything from six hours to several days work can be saved. The need for heavy lifting tackle is dispensed with and for some work, for example, clutch maintenance, it is not even necessary to drain off the cooling water and engine oil.

This split-chassis arrangement would appear to be advantageous for operators using a number of similar units, and in particular for the armed services. It provides, in effect, a vehicle built up from two sub-assemblies, and it might be possible by cannibalisation to produce a serviceable machine from the undamaged sub-assemblies of two partly wrecked ones.

The following times for operations were determined from the test:

Splitting the chassis in preparation for clutch change	2 hr
Re-assembly after changing clutch	1½ hr
Replacing a petrol engine with a diesel unit and carrying out the necessary modifications	6 hr

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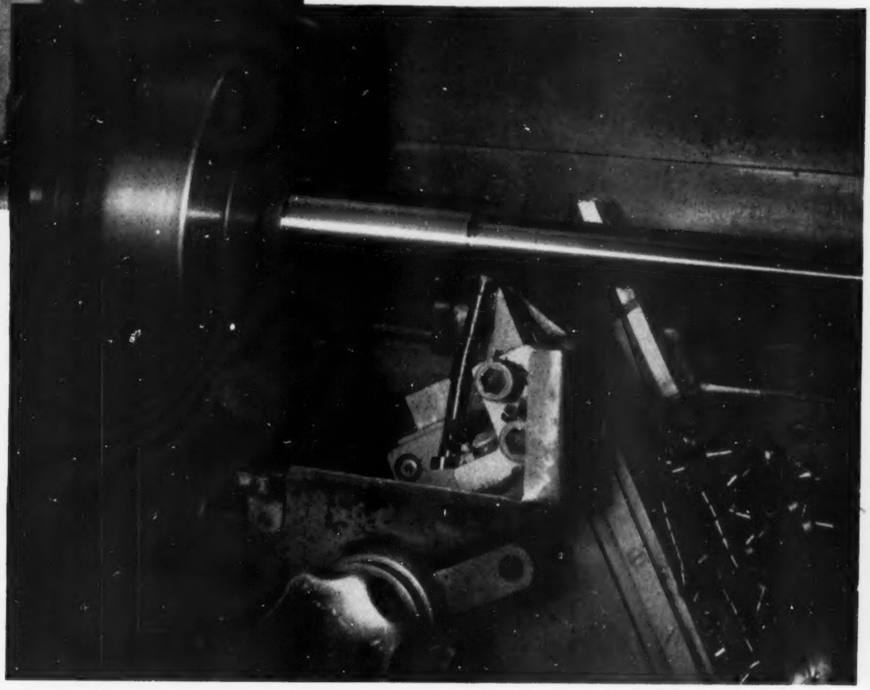

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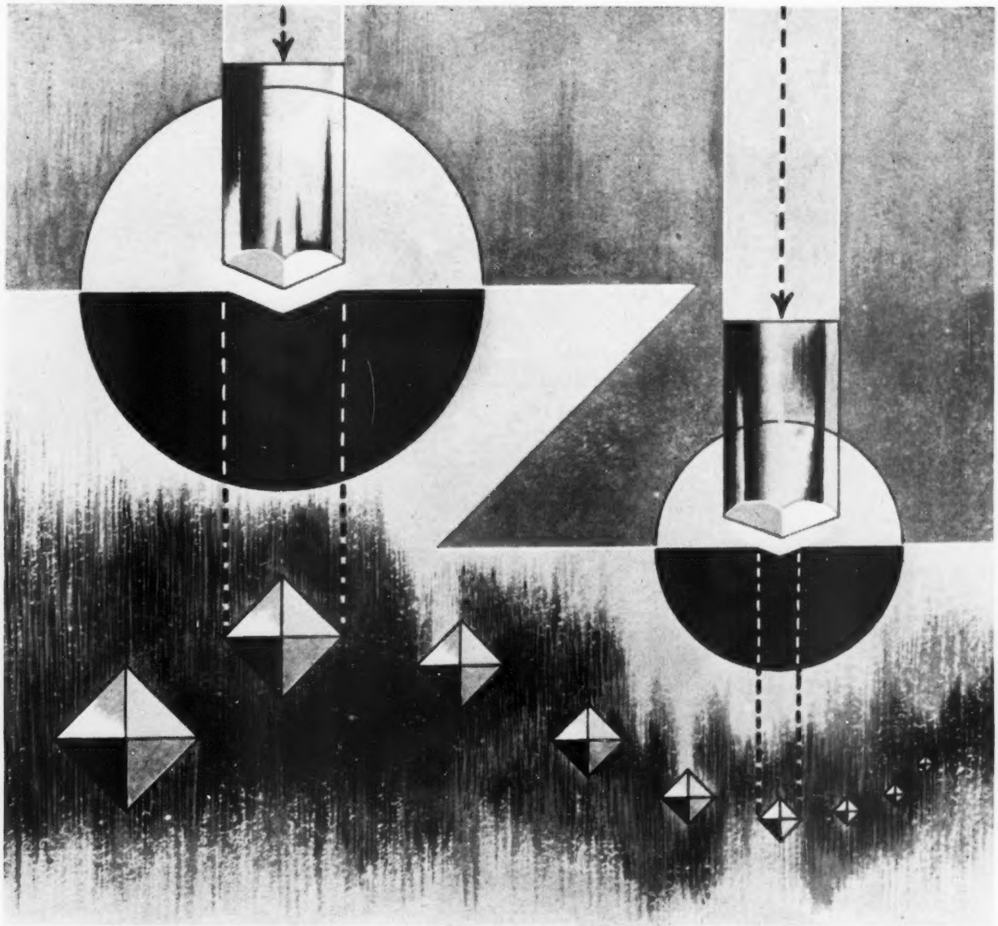

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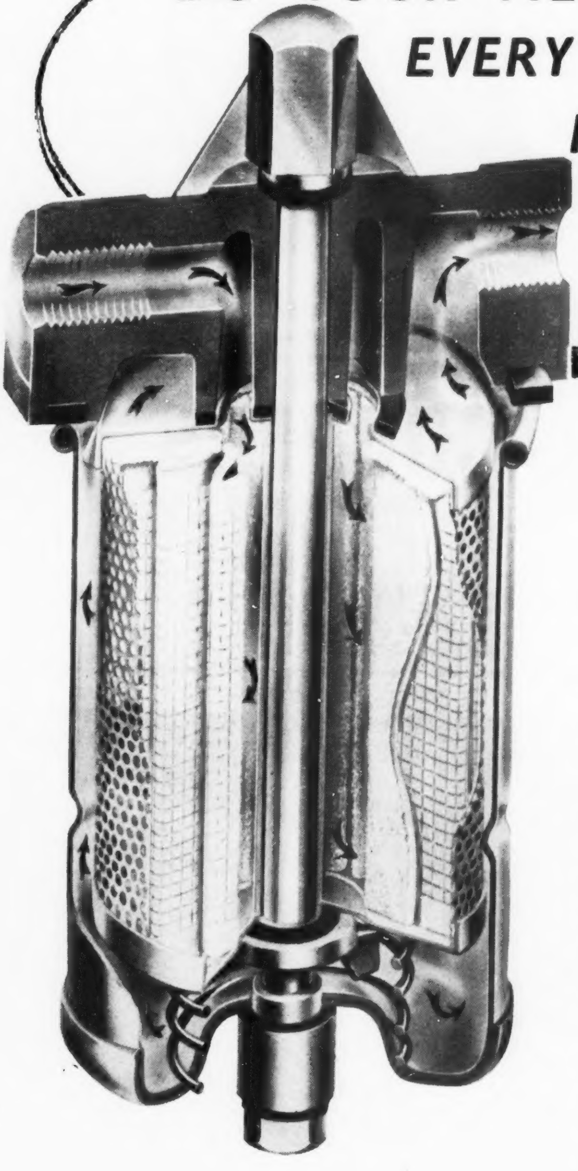

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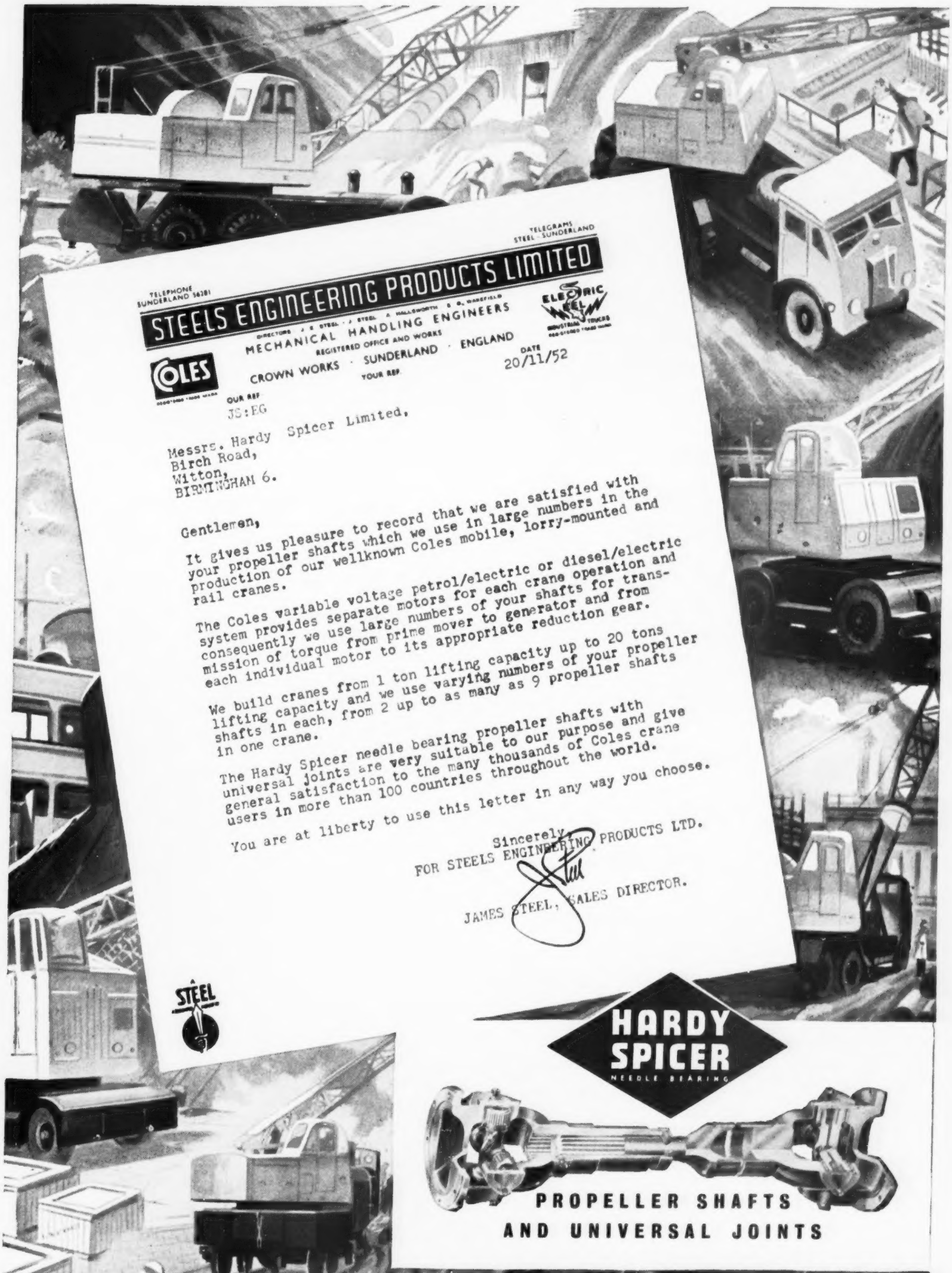

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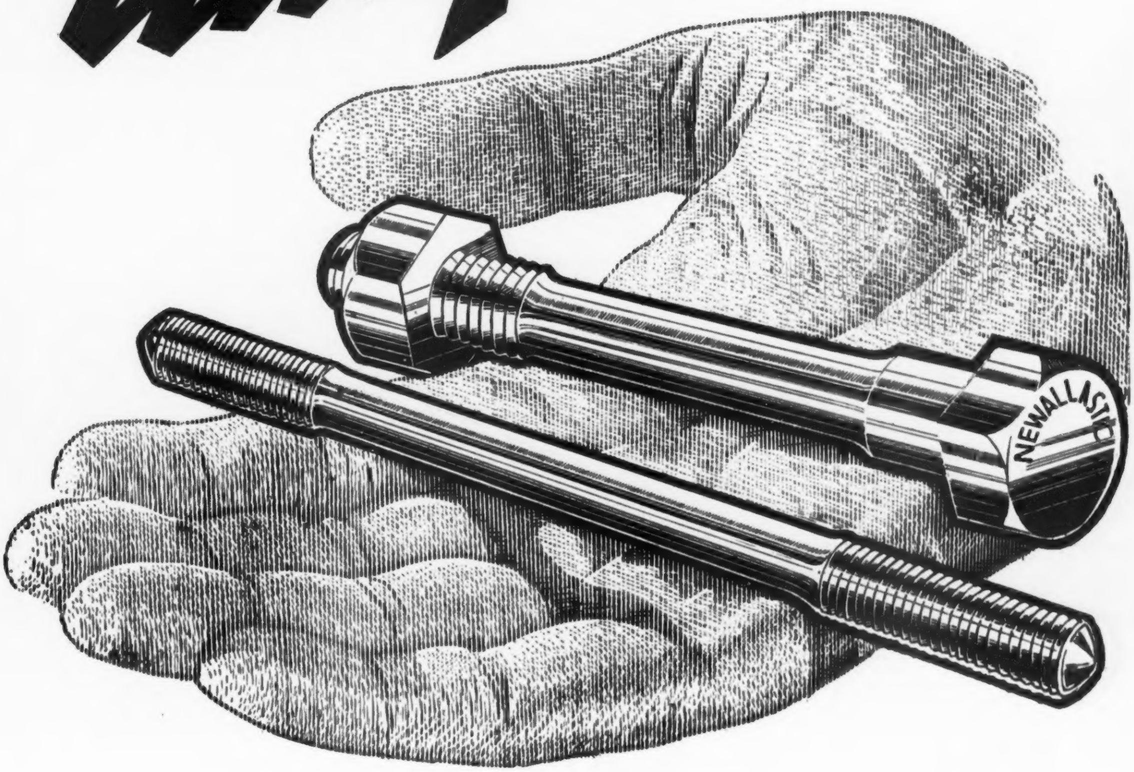

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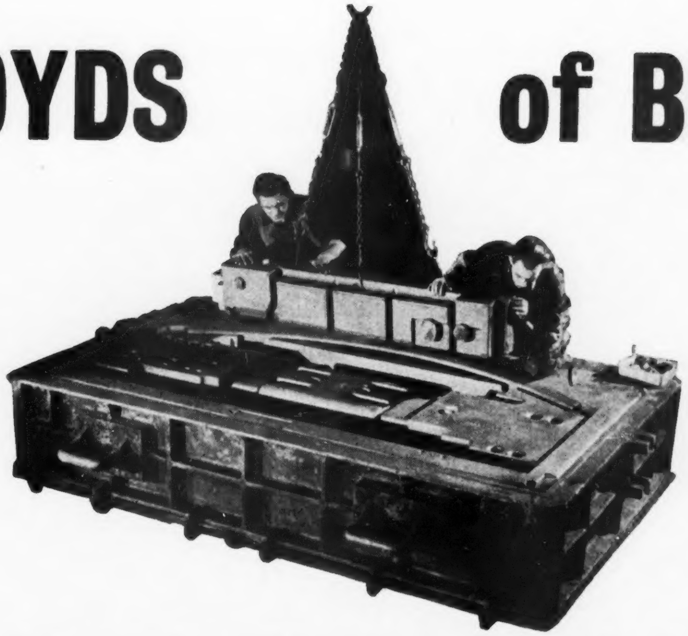

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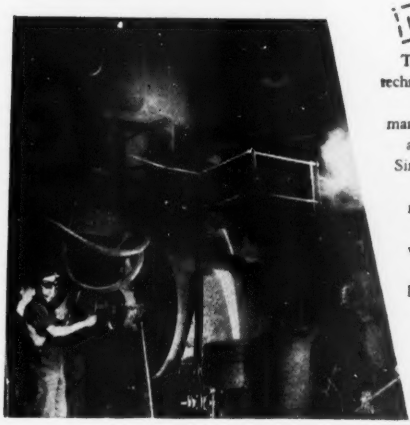

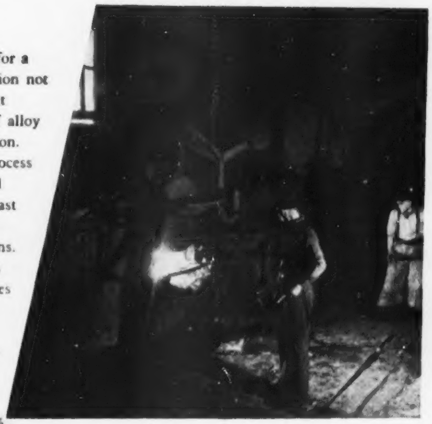

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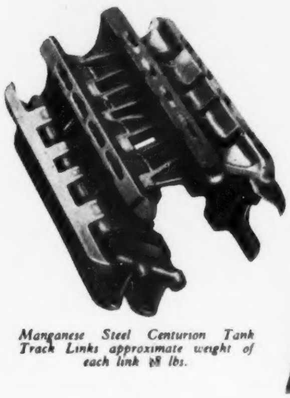

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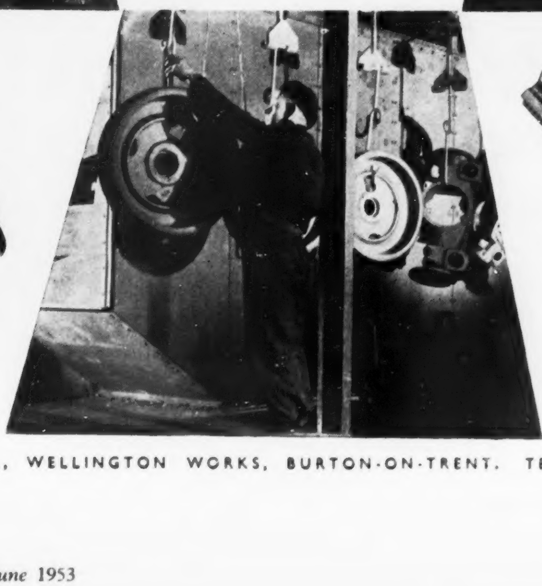

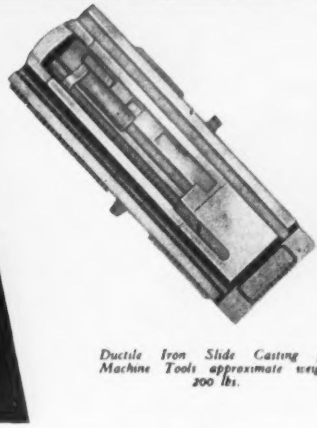

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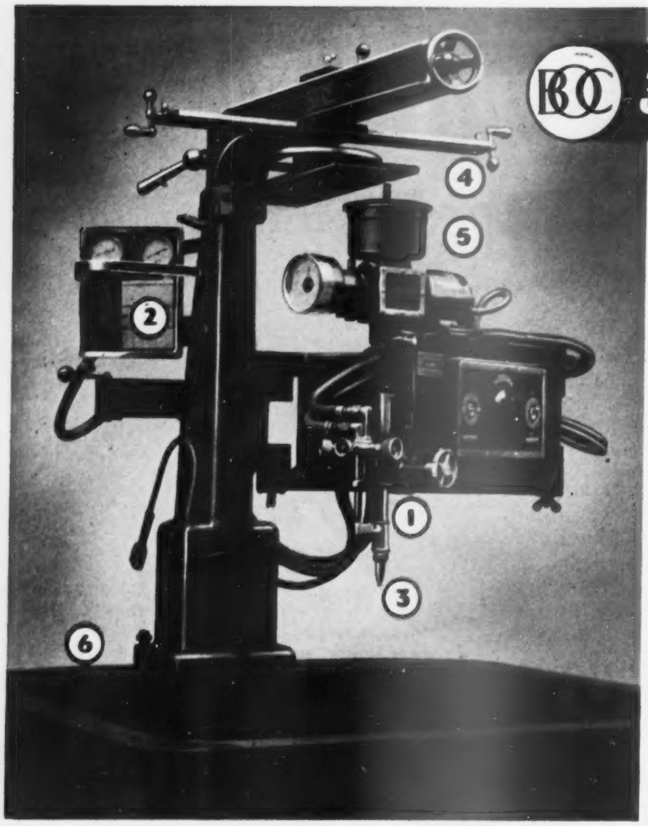

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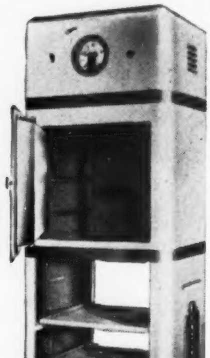

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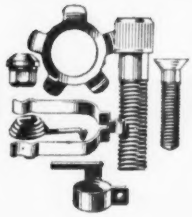

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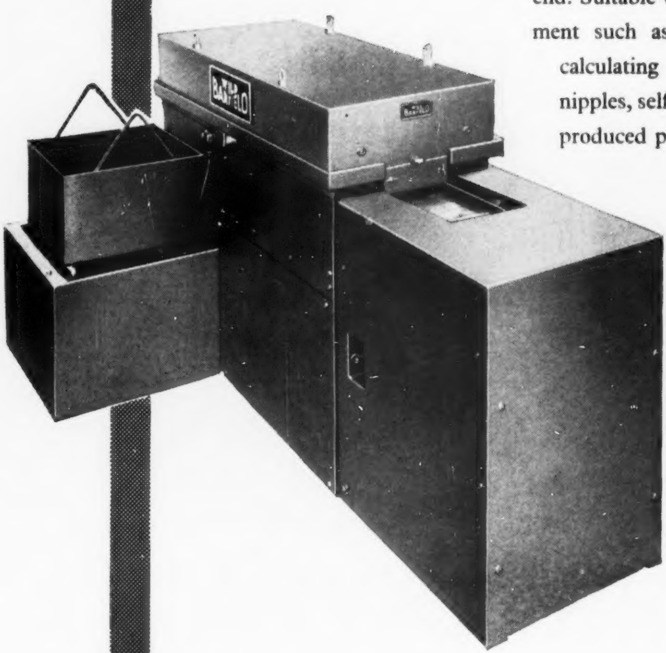

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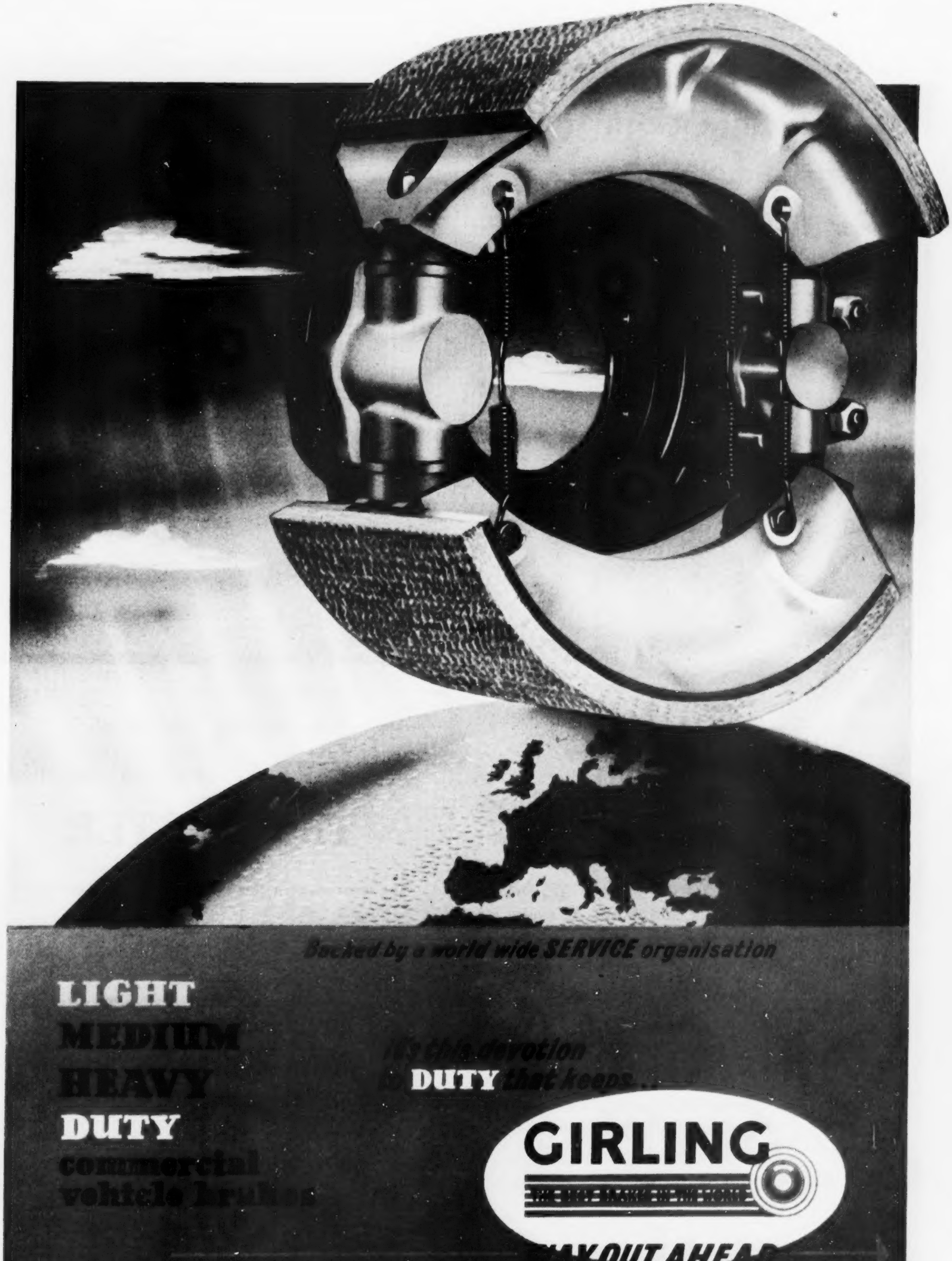

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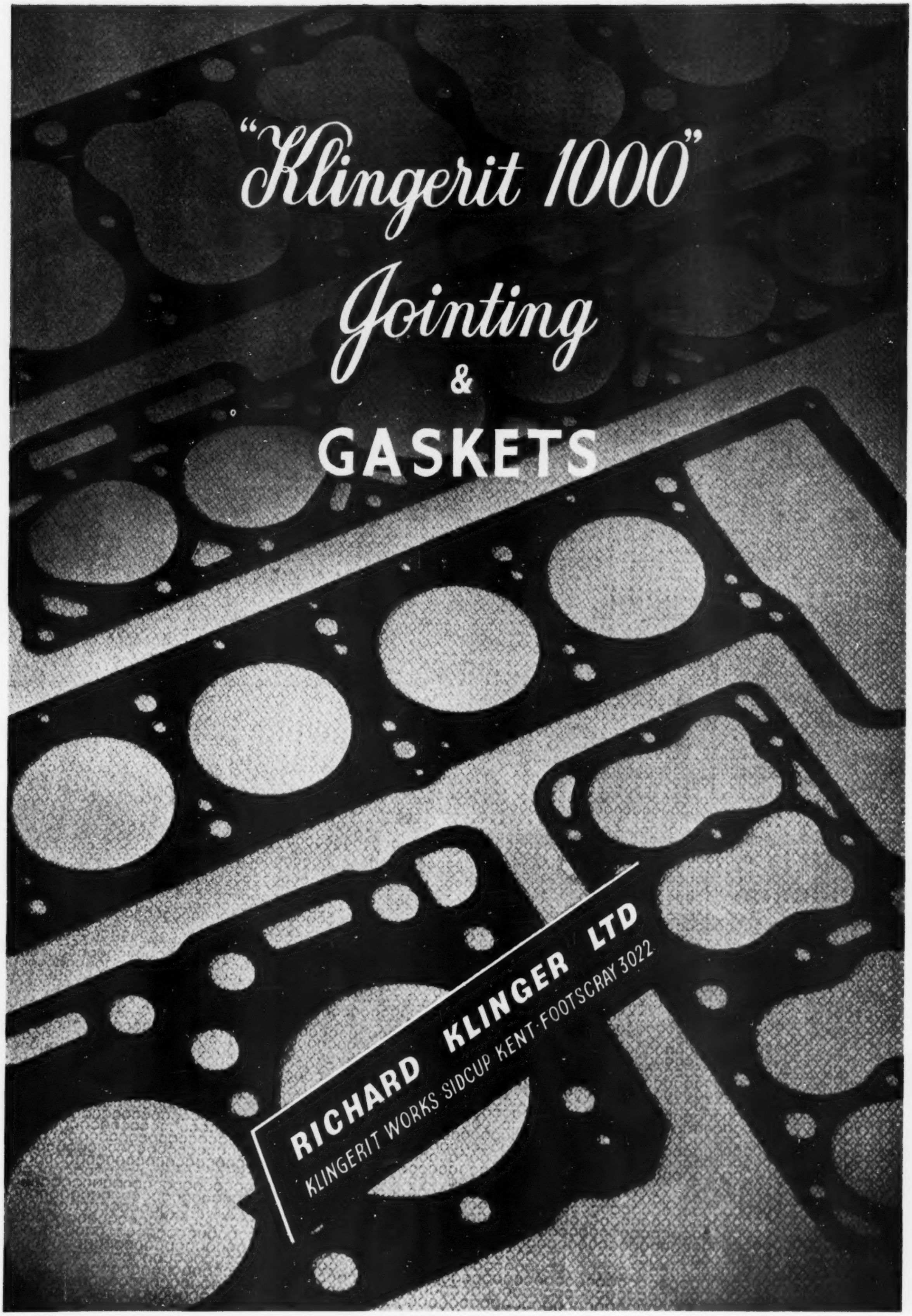
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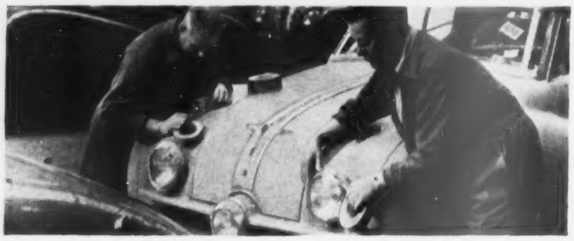

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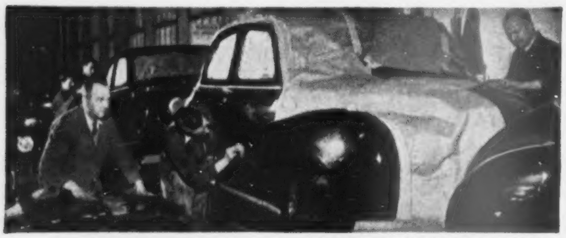

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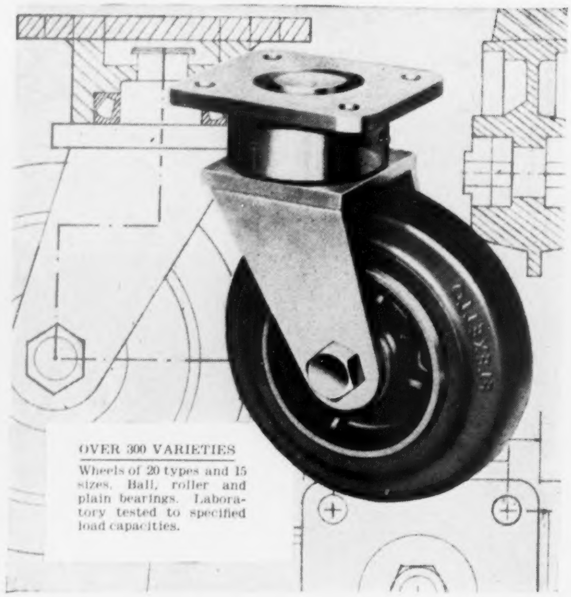

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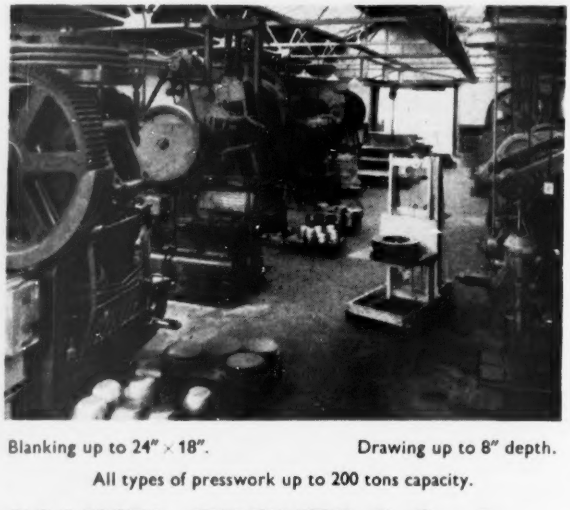

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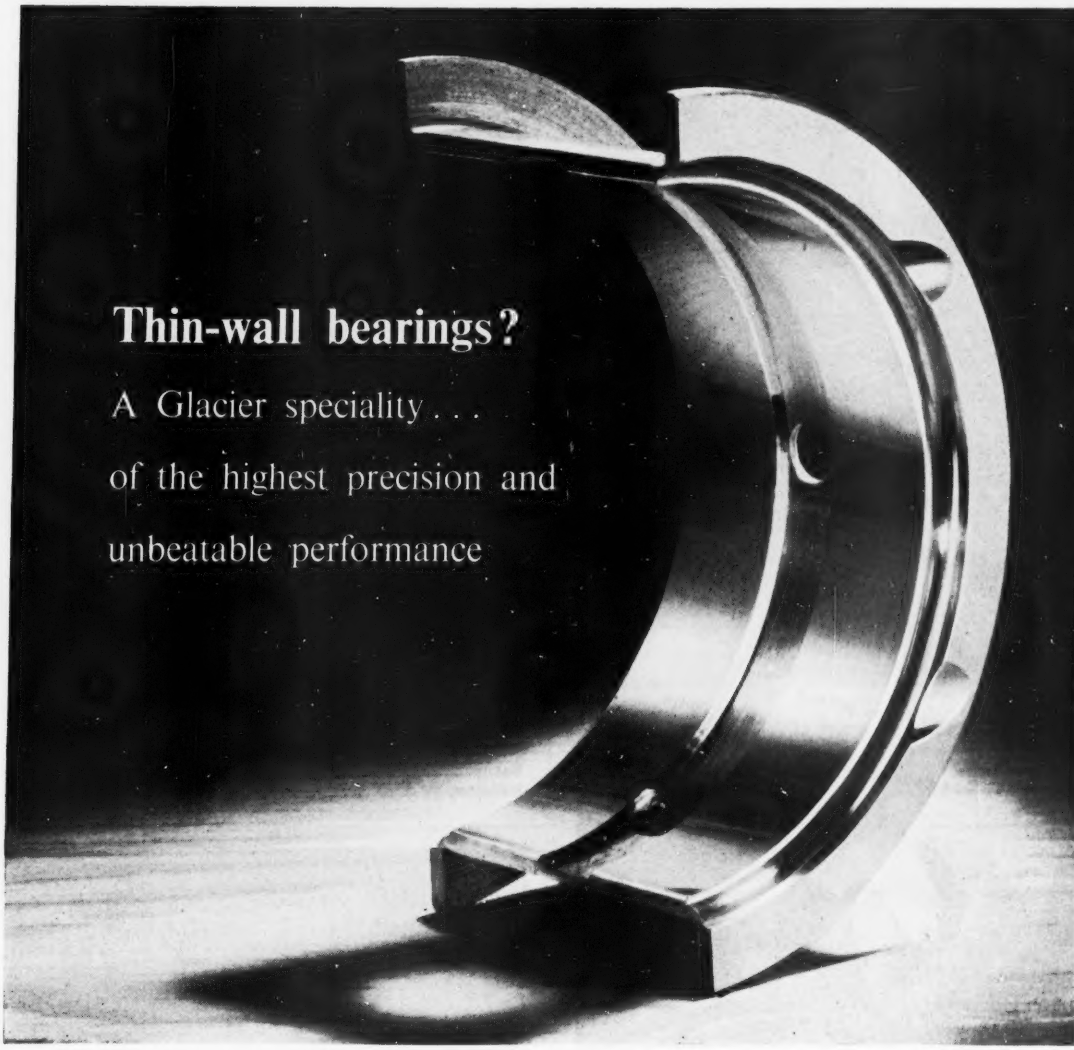

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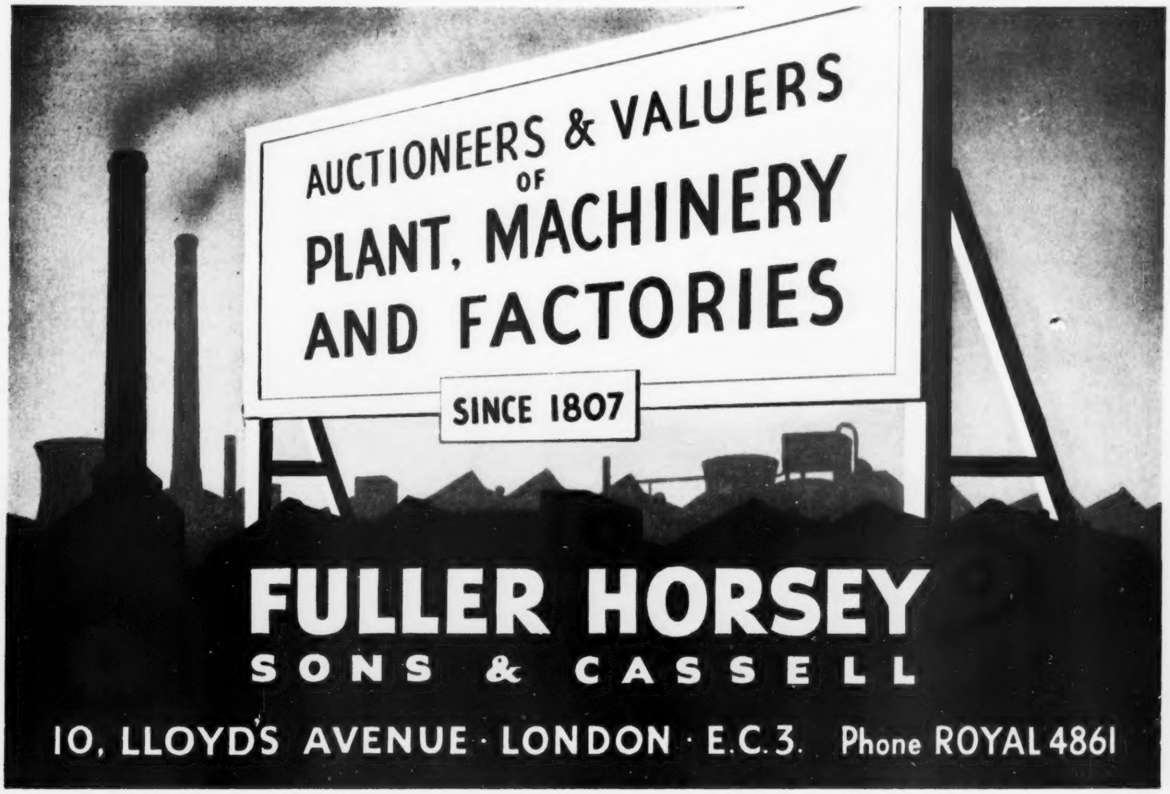

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